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ENERGY CONVERSION ALTERNATIVES STUDY

-ECAS-

GENERAL ELECTRIC PHASE I FINAL REPORT

VOLUME II, ADVANCED ENERGY CONVERSION SYSTEMS
Part 2, Closed Turbine Cycles

by

D.H. Brown, J.C. Corman, and R.B. Fleming

Corporate Research and Development
General Electric Company

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Project Manager, G.J. Barna
NASA Lewis Research Center
Cleveland, Ohio 44135

G.R. Fox, Program Manager
G.E. Corporate Research and Development
Schenectady, New York 12301

16. Abstract

A parametric study was performed to assist in the development of a data base for the comparison of advanced energy conversion systems for utility applications using coal or coal-derived fuels. Estimates of power plant performance (efficiency), capital cost, cost of electricity, natural resource requirements, and environmental intrusion characteristics were made for ten advanced conversion systems. Over 300 parametric points were analyzed to estimate the potential of these systems. Emphasis of the study was on the energy conversion system in the context of a base loaded utility power plant. Although cases employing transported coal-derived fuels were included in the study, the fuel processing step of converting coal to clean fuels was not investigated except for cases where a low-Btu gasifier was integrated with the power plant. All power plant concepts were premised on meeting emission standards requirements. The investigative approach focused on achieving consistency and comparability in the analysis of the various conversion systems. Recognized advocate organizations were employed to analyze their respective cycles and to present their analyses for power plant integration by the GE systems evaluation team. Wherever possible, common subsystems and components for the various systems were treated on a uniform basis. A steam power plant (3500 psig, 1000 F, 1000 F) with a conventional coal-burning furnace-boiler was analyzed as a basis for comparison. Combined cycle gas/steam turbine system results indicated competitive efficiency and a lower cost of electricity compared to the reference steam plant. The Open-Cycle MHD system results indicated the potential for significantly higher efficiency than the reference steam plant but with a higher cost of electricity. The information contained in this report constitutes results from the first phase of a two phase effort. In Phase II, a limited number of concepts will be investigated in more detail through preparation of conceptual designs and an implementation assessment including preparation of R&D plans estimating the resources and time required to bring the systems to commercial fruition.

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FOREWORD

The work described in this report is a part of the Energy Conversion Alternatives Study (ECAS)—a cooperative effort of the Energy Research and Development Administration, the National Science Foundation, and the National Aeronautics and Space Administration.

This General Electric contractor report for ECAS Phase I is contained in three volumes:

Volume I - Executive Summary

Volume II - Advanced Energy Conversion Systems

- Part 1 - Open-Cycle Gas Turbines
- Part 2 - Closed Turbine Cycles
- Part 3 - Direct Energy Conversion Cycles

Volume III - Energy Conversion and Subsystems and Components

- Part 1 - Bottoming Cycles and Materials of Construction
- Part 2 - Primary Heat Input Systems and Heat Exchangers
- Part 3 - Gasification, Process Fuels, and Balance of Plant

In addition to the principal authors listed, members of the technical staffs of the following subcontractor organizations developed information for the Phase I data base:

General Electric Company
- Advanced Energy Programs/Space Systems Department
- Direct Energy Conversion Programs
- Electric Utility Systems Engineering Department
- Gas Turbine Division
- Large Steam Turbine-Generator Department
- Medium Steam Turbine Department
- Projects Engineering Operation/I&SE Engineering Operation
- Space Sciences Laboratory

Actron, a Division of McDonnell Douglas Corporation

Argonne National Laboratory

Avco Everett Research Laboratory, Incorporated

Bechtel Corporation

Foster Wheeler Energy Corporation

Thermo Electron Corporation

This General Electric contractor report is one of a series of three reports discussing ECAS Phase I results. The other two reports are the following: Energy Conversion Alternatives Study (ECAS), Westinghouse Phase I Final Report (NASA CR-134941), and NASA Report (NASA TMX-71855).

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Summary

ADVANCED ENERGY CONVERSION SYSTEMS

The objective of Phase I of the Energy Conversion Alternatives Study (ECAS) for coal or coal-derived fuels was to develop a technical-economic information base on the ten energy conversion systems specified for investigation. Over 300 parametric variations were studied in an attempt to identify system and cycle conditions which indicate the best potential of the energy conversion concept. This information base provided a foundation for selection of energy conversion systems for more in-depth investigation in the conceptual design portion of the ECAS study. The systems for continued study were specified by the ECAS Interagency Steering Committee.

The technical-economic results include efficiency, capital cost and cost of electricity. For reference purposes a steam cycle (3500 psi/1000 F/1000 F [2.41×10^7 N/m^2/811 K/811 K]) with conventional coal burning furnace, stack gas cleanup and wet mechanical draft cooling towers was analyzed with the same analysis procedure employed for the advanced systems. This reference steam plant had an efficiency of approximately 37 percent. The open-cycle MHD system was the only plant to show efficiencies approaching 50 percent. A group of cycles-advanced steam, supercritical CO_2, liquid metal topping, and inert gas MHD— were estimated to have efficiencies in the 40 to 45 percent range.

The energy conversion systems with capital costs significantly lower than the reference steam plant were those with short construction times and simple construction, i.e., open-cycle gas turbines and low-temperature fuel cells. The more complex plants, i.e., open- and closed-cycle MHD and liquid metal topping, required longer construction time and were higher in capital cost.

Efficiency and capital cost are a part of the total technical-economic evaluation. The combination of these characteristics with the cost of fuel and operation and maintenance costs results in a cost of electricity for more complete comparisons. The only systems which were consistently lower than the reference steam plant's 30 mills/kWh at 65 percent capacity factor were the open-cycle gas turbine-combined cycle. MHD, supercritical CO_2, liquid metal top topping, and high-temperature fuel cells had a higher cost of electricity than the reference steam plant, as did many of the advanced steam cases because of their higher capital costs. The low capital cost plants— (low-temperature fuel cells and open cycle gas turbine, recuperative) utilized clean fuels and consequently had high fuel charges. These systems would be more economically applicable to peaking or mid-range duty.

ADVANCED ENERGY CONVERSION SYSTEMS

Many advanced energy conversion techniques which can use coal or coal-derived fuels have been advocated for power generation applications. Conversion systems advocated have included open- and closed-cycle gas turbine systems (including combined gas turbine-steam turbine systems), supercritical CO_2 cycle, liquid metal Rankine topping cycles, magnetohydrodynamics (MHD), and fuel cells. Advances have also been proposed for the steam systems which now form the backbone of our electric power industry. These advances include the use of new furnace concepts and higher steam turbine inlet temperatures and pressures. Integration of a power conversion system with a coal processing plant producing a clean low-Btu gas for use in the power plant is still another approach advocated for energy conserving, economical production of electric power. Studies of all these energy conversion techniques have been performed in the past. However, new studies performed on a common basis and in light of new national goals and current conditions are required to permit an assessment of the relative merits of these techniques and potential benefits to the nation.

The purpose of this contract is to assist in the development of an information base necessary for an assessment of various advanced energy conversion systems and for definition of the research and development required to bring these systems to fruition. Estimates of the performance, economics, natural resource requirements and environmental intrusion characteristics of these systems are being made on as comparable and consistent a basis as possible leading to an assessment of the commercial acceptability of the conversion systems and the research and development required to bring the systems to commercial reality. This is being accomplished in the following tasks:

Task I	Parametric Analysis	(Phase I)
Task II	Conceptual Designs	(Phase II)
Task III	Implementation Assessment	(Phase II)

This investigation is being conducted under the Energy Conversion Alternatives Study (ECAS) under the sponsorship of Energy Research and Development Administration (ERDA), National Science Foundation (NSF), and National Aeronautics and Space Administration (NASA). The control of the program is under the direction of an Interagency Steering Committee with participation of the supporting agencies. The NASA Lewis Research Center is responsible for project management of this study.

The information presented in this report describes the results produced in the Task I portion of this study. The emphasis

in this task was placed upon developing an information base upon which comparisons of Advanced Energy Conversion Techniques using coal or coal-derived fuels can be made. The Task I portion of the study was directed at a parametric variation of the ten advanced energy conversion systems under investigation. The wide-ranging parametric study was performed in order to provide data for selection by the Interagency Steering Committee of the systems and specific configurations most appropriate for Task II and III studies.

The Task II effort will involve a more detailed evaluation of seven advanced energy conversion systems and result in a conceptual design of the major components and power plant layout. The Task III effort will produce the research and development plans which would be necessary to bring each of the seven Task II systems to a state of commercial reality and then to assess their potential for commercial acceptability.

A prime objective of this study was to produce results which had a cycle-to-cycle consistency. In order to accomplish this objective and still ensure that each system was properly advocated, an organization which is or had been a proponent of the prime cycle was selected to advocate the energy conversion system and to analyze the performance and economics of the prime cycle portion of the energy conversion system, i.e., the parts of the system which were novel or unique to the system. The remaining subsystems, e.g., fuel processing, furnaces, bottoming cycles, balance of plant, were analyzed by technology specialist organizations which presently have responsibility for supplying these subsystems for utility applications. The final plant configuration and performance were produced by the General Electric Corporate Research and Development study team and this group performed the critical integration of the final plant concept. This methodology was used to provide a system-to-system consistency while maintaining the influence of a cycle advocate.

The ten energy conversion systems under investigation in this study are defined and analyzed in this volume of the report. These include:

1. Open-cycle Gas Turbine Recuperative
   - with clean and semi-clean fuels produced from coal
   - with and without organic bottoming cycles
2. Open-Cycle Gas Turbine
   - with air and water cooling of the gas turbine hot gas path
   - with clean and semi-clean fuels from coal and integrated low-Btu gasifiers

3. Closed-Cycle Gas Turbine

   - with helium working fluid
   - with a variety of direct coal and clean fuel furnaces
   - with and without organic and steam bottoming cycles

4. Supercritical CO_2 Cycle

   - with basic and recompression cycle variations
   - with a variety of direct coal and clean coal-derived fuel furnaces

5. Advanced Steam Cycle

   - with both throttle and/or reheat temperatures greater than present practice (1000 F [811 K])
   - with a variety of direct coal and clean coal-derived fuel furnaces

6. Liquid Metal Topping Cycle

   - with potassium and cesium as working fluids
   - with a variety of direct coal and clean fuel furnaces

7. Open-Cycle MHD

   - with direct coal and semi-clean fuel combustion
   - with standard steam and gas turbine bottoming

8. Closed-Cycle Inert Gas MHD

   - with parallel and topping configurations
   - with both direct coal and semi-clean fuel utilization

9. Closed-Cycle Liquid Metal MHD

   - with mixture of liquid sodium and helium as working fluids
   - with standard steam bottoming
   - with a variety of direct coal and clean fuel furnaces

10. Fuel Cells

   - both high and low temperature (less than 300 F [422 K])
   - with employment of clean process fuels for low temperature cells and low-Btu gasification at high temperature cells

The subsystems which complete the energy conversion system are discussed in Volume III of this report. The results as presented in the following sections include the total energy conversion system.

2.4 CLOSED-CYCLE GAS TURBINE

DESCRIPTION OF CYCLE

The schematic for the closed-cycle gas turbine is presented in Figure 2.4-1 with atmospheric fluidized beds (AFB) burning coal, capturing sulfur, and heating the helium. The highly effective recuperator results in a helium temperature of 875 F (741.5 K) entering the AFB. As a result the combustion gases are cooled to only 1000 F (811 K) in the AFB, and a high-temperature air preheater is needed to cool the exhaust to the 730 F (661 K) level for electrostatic precipitation of solids. The low-temperature air preheater brings the stack gas to 300 F (422 K) to achieve minimal stack loss. The wet cooling towers service the precooler where the helium temperature is reduced from 463 F (513 K) to 80 F (300 K) at the compressor inlet. The compressor inlet flow was 1031 lb/s (467.6 kg/s) in every case, and the compressor discharge pressure was 1000 psia (6.9 MN/m^2). Blocking helium coolant flows were used for the turbine, but the nozzles and buckets were not cooled.

When a bottoming cycle was added, the organic boiler or the steam boiler substitutes for a part of the temperature range of the recuperator and the precooler. The reduction of heat added to the compressed helium from the recuperator results in a lower helium temperature entering the AFB. The AFB design for such cases was changed to take full advantage of the cooler helium to reduce both AFB size and cost.

In addition to the AFB configuration, the primary heat input was also evaluated for a pressurized fluidized bed serviced by a gas turbine with an exhaust gas recuperator. Clean gaseous fuels were evaluated for use in pressure fired furnaces to heat the helium. The pressurizing gas turbines had exhaust heat recovery steam generators and a steam turbine. These units were integrated with a gasification plant when the clean fuel was low-Btu gas and not over-the-fence high-Btu gas.

ANALYTICAL PROCEDURES AND ASSUMPTIONS

All helium cycles had a compressor inlet flow of 1031 lb/s (467.6 kg/s) of helium; the compressor discharge pressure in every case was 1000 psia (6.9 MN/m^2). The overall pressure ratio was achieved by variation of the pressure level for the section from the turbine outlet to the compressor inlet. Overspeed control valves, separate from the combustion system temperature control, may be required to allow for sudden load loss. These valves would be bypass valves providing a flow path between the compressor discharge and the turbine exhaust. Control valve leakage would be approximately 0.2 percent of total compressor flow. No provision was made for this parasitic leakage flow in these evaluations.

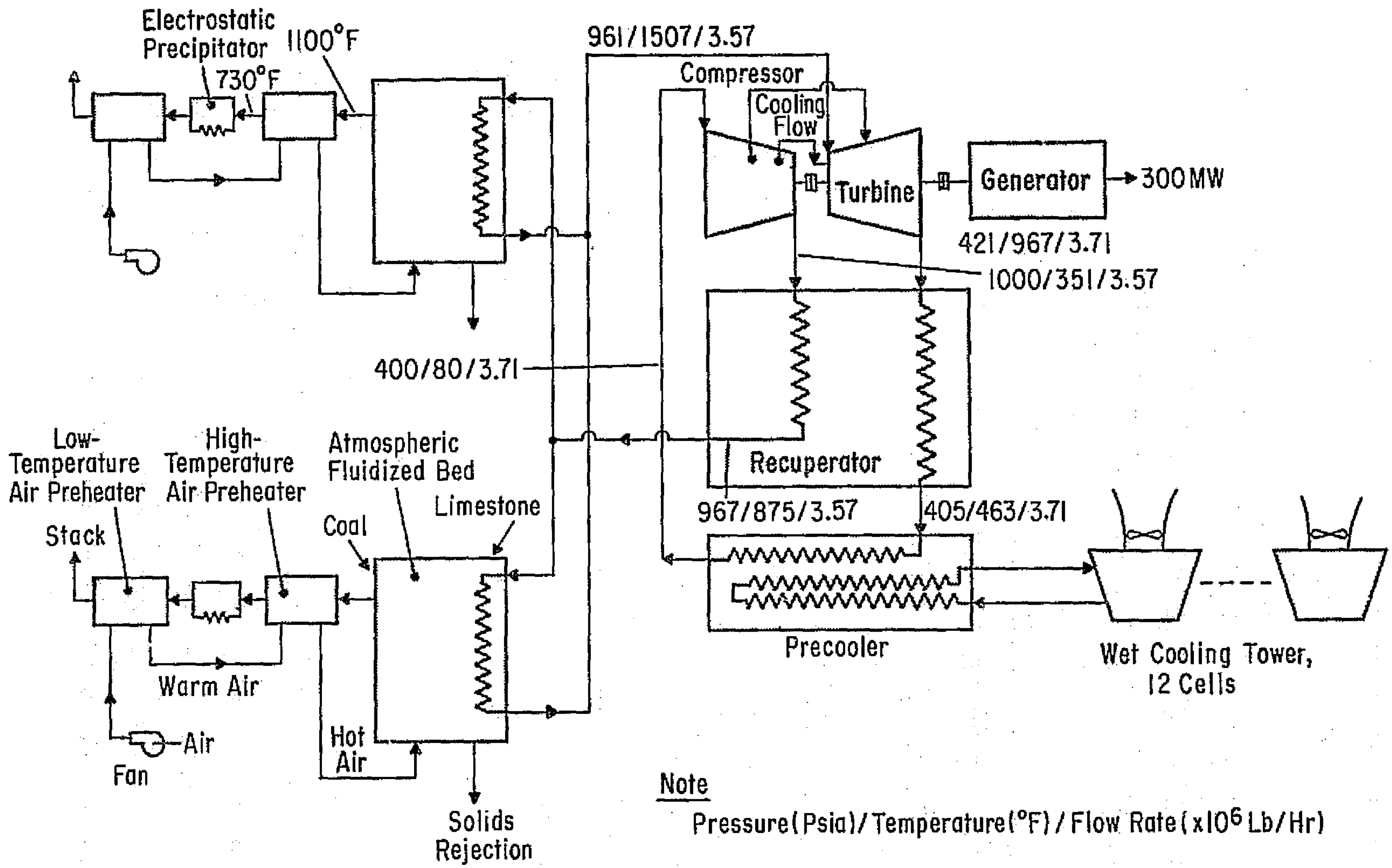

Figure 2.4-1. Closed-Cycle Gas Turbine

Cycle Components

Compressor. Helium compressors have been designed for use in closed-loop helium cycles for the high-temperature gas-cooled reactor (HTGR) with pressure ratios within the range under consideration. Peak cycle pressures at the compressor discharge are 1000 psia (6.9 MN/m^2) or less. A single polytropic compressor efficiency has been assumed for this study. The resulting adiabatic efficiency is a function of pressure ratio and is the "blading efficiency" measured from the inlet total pressure and total temperature ahead of the inlet guide vanes to the total temperature and total pressure downstream of the exit guide vanes. The compressor discharge diffuser pressure loss was considered part of the overall ducting system. Turbine cooling flows are bled from the compressor at three interstage locations as well as at compressor discharge. The compressor RPM was 3600 for all cases; the inlet flow was 1031 lb/s (467.6 kg/s) and the inlet temperature was 80 F (300 K).

Turbine. Five-, six-, and seven-stage turbines were required for efficient utilization of compressor pressure ratios of 2.0, 2.5, and 3.0, respectively. Turbine efficiency was calculated as the blading total-to-total adiabatic efficiency. Inlet and exhaust duct losses are considered part of the piping system pressure loss. The assigned turbine stage adiabatic efficiency was fixed for the shrouded stages, and set at a slightly reduced value for the unshrouded stages. Coolant flows were treated as merging with the main helium flow immediately behind the stage that the coolant cooled.

Heat Source. The helium flow through the heat source was assigned a pressure loss of 1.5 percent of the absolute pressure level. The details of the furnace configurations and the heating surface deployment are found in Section 6.

Recuperator. Although HTGR cycle studies have shown recuperators of 89.5 percent effectiveness, to date such units have not been constructed. A more conservative value of 85 percent effectiveness was designated for the base case, with variations to 90 percent and 95 percent for parametric points. The assigned helium flow pressure losses were 2 percent on the low-pressure side and 1 percent on the high-pressure side. The sizing and materials selected for the recuperator are detailed in Section 7.

Precooler. Waste heat is rejected from the cycle to cooling water through the precooler. The precooler is a straight-tube, axial, counterflow heat exchanger. Heat is transferred from helium to water. Helium pressure loss for the precooler was 1 percent. The precooler and cooling towers can provide for a compressor inlet temperature which is 20 F (11.1 K) above the ambient air temperature.

Gas Properties for Helium. Helium was treated as an ideal gas for the pressures and temperatures of this study in accordance with the properties tabulated in the National Bureau of Standards Circular 564.

Bottoming Cycle Components

The substitution of steam or organic fluid boilers for parts of the recuperator and precooler was deemed to impose the same helium flow pressure loss as the unit replaced. The assumptions for these cycles have been detailed in Section 7 for their heat exchangers and Section 4 for their cycle components and configurations.

Flow Pressure and Other Losses

The pressure losses through the connecting ducts and pipe work were appraised in detail, with a resulting value of 4.3 percent for these additional losses. In combination with the losses already enumerated, the total helium circuit would have a loss of 8.733 percent. The result of this loss is that the turbine pressure ratio for expanding the helium is 91.267 percent of the compressor pressure ratio.

The generator was assigned an efficiency appropriate for a large hydrogen-cooled machine. The losses were excitation, windage, bearings, and seals, as well as electrical losses. The mechanical and accessory losses for the turbine and compressors were assigned at 0.3 percent of the generator output.

DESIGN AND COST ANALYSIS

Materials appropriate for the helium gas turbine are presented in Table 2.4-1.

These materials are all currently in use and would be applied within known property limits. The hot gas path parts may have coatings or claddings applied to increase their endurance. The cost basis for the helium gas turbine was determined by detailed consideration of the base case unit. Thereafter differentials were determined for the effects of pressure ratio, inlet temperature, and generator output. Included in the gas turbine unit cost was provision for 100 ft (30.5 m) of helium ducting from the heat source to the turbine.

The recuperators were sized using 1-in. diameter tubes of stainless steel as described in Section 7. Tube sheets were of one-half Chrome, one-half Moly up to 1000 F (811 K), and were of stainless steel above 1000 F. Recuperator shells were of one-half Chrome, one-half Moly to 1000 F; and of two and one-fourth Chrome, one Moly above 1000 F. As a result of these specifications there were distinct steps in recuperator cost when turbine exhaust exceeded 1000 F.

Table 2.4-1

MATERIAL SELECTIONS FOR HELIUM GAS TURBINE

Components	Alloy Selected	Alternate Alloy
Turbine		
Buckets	M-21 LC	René-100, Mo-T2M
Nozzles	M-21 LC	IN-713 LC, Mo-T2M
Wheels	IN-706	A-286, M-152
Casings	HAST X	
Inlet pipe	IN-713 LC	
Exhaust pipe	Cr Mo	
Compressor		
Rotor blades	403 SS	
Stator blades	403 SS	
Wheels	Ni Cr Mo V	MS-250

The organic bottoming design and costs were identical to the basis outlined for the recuperative gas turbine bottoming. The bottoming steam turbine and its heat recovery steam generator (HRSG) were comparable to the prior design basis as described in Section 4. The dry cooling tower designs and costs for the helium closed-cycle power plants differ from those for other power plants. The cooling water from the precooler may be as hot as 340 F (444 K), in contrast to approximately 120 F (322 K) from other closed-cycle plants.

RESULTS

The base case results are summarized in detail in Table 2.4-2. The 300 MW generator output was reduced to 276 MW net station output by the auxiliary demands of the furnace, the cooling towers, and the station services. The resulting overall energy efficiency was 29.5 percent. The plant cost in dollars per kilowatt is high compared with a steam plant, with major components contributing an appreciable share of the total. The environmental intrusions are comparable to other plants with atmospheric fluidized bed coal combustion.

The parametric point variables and results are presented in Table 2.4-3, with the companion capital cost distributions in Table 2.4-4, and the power generation and consumption detailed in Table 2.4-5. A single helium gas turbine was the basis for all cases except for Cases 9 and 10, with two and four helium gas turbines, respectively. These latter cases proved less economic than the base case because of the extension of the site

Table 2.4-2

SUMMARY SHEET
CLOSED-CYCLE GAS TURBINE BASE CASE

CYCLE PARAMETER

Net Power Output (MWe)	27[illegible]
Furnace and Coal Type	Atmospheric fluidized bed Illinois No. 6
Prime Cycle	
Turbine inlet temperature (°F)	1500
Compressor pressure ratio	2.5
Recuperator effectiveness	0.85
Recuperator pressure drop ($\Delta p/p$)	0.03
Loop ($\Delta p/p$)	0.08733
Compressor Inlet Temperature (°F)	80
Heat Rejection	Wet cooling tower

MAJOR COMPONENT CHARACTERISTICS

Major Component	Unit or Module: Size (ft) (W x L (or D) x H)	Weight (lb) (x 10^6)	Cost ($ x 10^6)	Units Required	Total Cost ($ x 10^6)	$/kW Output
Helium turbine-compressor-generator	16 x 66 x 25	1.36	14.7	1	14.7	53.3
Recuperator	24 x 45 x 7	0.2	0.8	20	15.0	54.3
Furnace modules	13 x 31 x 198	4.32	22.6	1.84	41.6	166.7
High-temperature air preheater	30 x 32 x 5	0.09	0.12	16.56	2.0	7.2

PERFORMANCE AND COST

Thermodynamic efficiency (percent)	36.4
Powerplant efficiency (percent)	29.5
Overall energy efficiency (percent)	29.[illegible]
Plant capital cost ($ x 10^6)	225
Plant capital cost ($/kWe)	814
Cost of electricity (mills/kWh)	38.8

NATURAL RESOURCES

Coal (lb/kWh)	1.07
Water (gal/kWh)	
Total	0.58
Cooling	0.58
Processing	0
Makeup	0
NO_x suppression	0
Stack gas cleanup	0
Land (acres/100 MWe)	12.0

ENVIRONMENTAL INTRUSION

	Lb/10^6-Btu Input	Lb/kWh Output
SO_2	1.084	1.25×10^{-2}
NO_x	0.274	0.32×10^{-2}
HC	--	
CO	0.214	0.25×10^{-2}
Particulates	0.100	0.11×10^{-2}

	Btu/kWh
Heat to water	5962
Heat, total rejected wastes	8154

	Lb/kWh	Lb/Day
Wastes		
Furnace solids	0.190	1.261×10^6
Fine dust from cyclones	0.141	0.936×10^6
Fly ash	0.020	0.132×10^6

FOLDOUT FRAME

PARAME

Parameters	Case 1*	2	3	4	5	6	7	8	9	10	11	12	13	14	15
Power Output (MWe)	276	275	276	732	789	771	335	398	552	1105	231	403	223	303	262
Furnace, Coal, and Conversion Process	AFB Ill. #6	AFB N.D.	AFB Mont	PF Ill. #6 LBtu	PF N.D. LBtu	PF Mont LBtu	PF Ill. #6 HBtu	$(PFB)_R$ Ill. #6	AFB Ill. #6 →	→	→	PF Ill. #6 HBtu	AFB Ill. #6 →	→	→
Prime Cycle															
Turbine inlet temperature (°F)	1500 →	→	→	→	→	→	→	→	→	→	1350	1700	1500 →	→	→
Compressor pressure ratio	2.5 →	→	→	→	→	→	→	→	→	→	→	→	2.0	3.0	2.5
Recuperator effectiveness	0.85 →	→	→	→	→	→	→	→	→	→	→	→	→	→	→
Recuperator pressure drop (Δp/p)	0.03 →	→	→	→	→	→	→	→	→	→	→	→	→	→	0.05
Loop (Δp/p)	0.03733 →	→	→	→	→	→	→	→	→	→	→	→	→	→	0.1061
Compressor inlet temperature (°F)	80 →	→	→	→	→	→	→	→	→	→	→	→	→	→	→
Heat rejection	WCT →	→	→	→	→	→	→	→	→	→	→	→	→	→	→
PF Air Supply															
Excess air (percent)	--	--	--	15 →	→	→	10	20	--	--	--	10	--	--	--
Pressure ratio	--	--	--	10 →	→	→	8	10	--	--	--	8	--	--	--
Furnace exit temperature (°F)	--	--	--	1600 →	→	→	1200	1600	--	--	--	1200	--	--	--
Regenerator efficiency	--	--	--	Steam →	→	→	--	0.85	--	--	--	--	--	--	--
Bottoming Cycle															
Turbine inlet temperature (°F)	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--
Superheat to inlet gas temperature differential (°F)	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--
Boiler pinch point temperature differential (°F)	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--
Heat rejection	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--
T condense (°F)	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--
Actual Powerplant Output (MWe)	276	275	276	732	789	771	335	398	552	1105	231	403	223	303	262
Thermodynamic Efficiency (percent)	36.4	36.4	36.4	36.4	36.4	36.4	36.4	36.4	36.4	36.4	34.5	30.4	33.7	34.3	35.2
Powerplant Efficiency (percent)	29.5	28.1	28.6	31.0	31.4	31.5	29.2	31.8	29.5	29.6	27.9	30.6	30.6	28.0	28.4
Overall Energy Efficiency (percent)	29.5	28.1	28.6	31.0	31.4	31.5	14.7	31.8	29.5	29.6	27.9	15.4	30.6	28.0	28.4
Coal Consumption (lb/kWh)	1.07	1.76	1.33	1.02	1.58	1.21	2.15	1.00	1.07	1.07	1.13	2.05	1.03	1.13	1.11
Plant Capital Cost ($ million)	225	228	226	507	575	515	152	300	475	1009	194	225	234	257	229
Plant Capital Cost ($/kWe)	814	829	821	691	728	660	454	753	859	912	839	559	1047	847	873
Cost of Electricity, Capacity Factor = 0.65															
Capital (mills/kWh)	25.7	26.2	26.0	21.9	23.0	21.1	14.4	23.8	27.2	28.8	26.5	17.1	31.1	26.8	27.4
Fuel (mills/kWh)	9.8	10.3	10.1	9.4	9.2	9.7	30.4	9.1	9.8	9.8	10.4	29.0	9.5	10.4	10.2
Maintenance and operating (mills/kWh)	3.2	3.2	3.2	3.3	3.0	3.3	1.9	3.0	2.2	1.7	3.3	1.6	3.9	3.1	3.4
Total (mills/kWh)	38.8	39.7	39.3	34.5	35.3	33.6	46.7	35.9	39.2	40.4	40.2	48.3	44.5	40.7	41.3
Sensitivity															
Capacity factor = 0.50 (total mills/kWh)	47.4	48.6	48.0	42.0	43.1	41.0	51.6	44.0	48.0	49.5	49.1	54.0	57.6	49.2	50.6
Capacity factor = 0.80 (total mills/kWh)	33.3	34.2	33.8	29.8	30.4	29.1	41.6	30.9	33.7	34.6	34.6	44.6	39.5	34.6	35.4
Capital Δ = 20 percent (Δmills/kWh)	5.1	5.2	5.2	4.4	4.6	4.2	2.9	4.8	5.4	5.8	5.3	3.5	6.6	5.4	5.5
Fuel Δ = 20 percent (Δmills/kWh)	2.0	2.1	2.0	1.9	1.8	1.8	6.1	1.8	2.0	2.0	2.1	5.8	1.9	2.1	2.0
Estimated Time for Construction (years)	4	4	4	4	4	4	3	4	5	6	4	3	4	4	4
Estimated Date of 1st Commercial Service (year)	1987	1987	1987	1987	1987	1987	1987	1987	1987	1987	1985	1989	1987	1987	1987

*Base case -- one gas turbine and one bottoming turbine where noted, except cases 9 and 10.
**Intercooled compressor

AFB = Atmospheric fluidized bed
DCT = Dry cooling tower
HBtu = High Btu
Ill. = Illinois
LBtu = Low Btu
Mont = Montana
N. D. = North Dakota
PF = Pressurized furnace
$(PFB)_R$ = Pressurized fluidized bed (
WCT = Wet cooling tower

Table 2.4-3

RIC VARIATIONS FOR TASK I STUDY
CLOSED-CYCLE GAS TURBINE

16	17	18	19	20	21	22	23	24	25	26	27	28	29	30	31	32	33	34	35	36	37	38	39	40	41	42	43	44	45	46
240	239	256	290	388	409	267	278	448	285	187	251	223	277	304	324	402	447	344	342	340	338	333	340	408	337	311	304	340	290	528
								PF Ill. #6 HBtu		AFB Ill. #6					PF Ill. #6 HBtu			AFB Ill. #6												
								1700		1350		1500			1700			1500												
			**	4**	6.25**	2.5		3	2		3	2	2.5	3	2	2.5	3	2.5												
							0.95	0.85				0.9						0.85						0.6	0.9	0.85		0.0	0.0	0.0
0.07	0.03																													
.1250	0.03333		0.1056			0.00733																								
	60	110	80			80	80																							
						DCT	WCT																DCT	WCT			DCT	WCT		
--	--	--	--	--	--	--	--	10		--	--	--	--	--	10			--	--	--	--	--	--	--	--	--	--	--	--	--
--	--	--	--	--	--	--	--	8		--	--	--	--	--	8			--	--	--	--	--	--	--	--	--	--	--	--	--
--	--	--	--	--	--	--	--	1200		--	--	--	--	--	1200			--	--	--	--	--	--	--	--	--	--	--	--	--
--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--
																					Organic							Steam		
--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	430	410	390	410	390	430	460	400	413		561	384	600
--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	33	53	73	53	73	33	151	34	60				67
--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	30			50	70	30	50	30					
--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	WCT					DCT	WCT			DCT	WCT		
--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	92					99.1	92			99.1	92		
244	239	256	298	388	409	267	278	448	285	187	251	223	277	304	324	402	447	344	342	340	338	333	340	408	337	311	304	340	290	528
33.9	37.6	34.4	37.4	38.3	35.0	35.8	40.3	38.2	35.0	33.0	33.7	37.0	38.2	37.4	37.6	39.8	39.3	44.7	44.4	44.1	44.0	43.3	44.2	42.6	45.7	40.7	39.9	35.9	40.9	37.6
27.3	30.6	27.8	30.5	31.6	28.8	28.6	32.8	30.4	24.5	26.4	27.4	29.9	31.1	30.5	29.9	31.5	31.1	36.0	36.5	36.3	36.2	35.5	36.4	35.3	37.8	33.3	32.5	29.5	33.4	30.8
27.3	30.6	27.8	30.5	31.6	28.8	28.6	32.8	15.3	12.4	26.4	27.4	29.9	31.1	30.5	15.1	15.9	15.7	36.8	36.5	36.3	36.2	35.5	36.4	35.3	37.8	33.3	32.5	29.5	33.4	30.8
1.16	1.03	1.14	1.04	1.00	1.10	1.11	0.96	2.07	2.56	1.20	1.15	1.06	1.07	1.04	2.10	1.99	2.02	0.86	0.87	0.87	0.87	0.89	0.87	0.90	0.84	0.95	0.97	1.07	0.95	1.03
225	233	230	238	274	341	260	319	239	230	171	216	305	251	277	258	238	240	326	326	326	323	321	326	352	334	288	288	312	310	420
907	894	896	796	706	832	973	1148	533	805	911	861	1368	906	910	797	591	558	947	954	960	955	964	957	862	993	924	946	918	1041	813
28.7	25.4	28.4	25.2	22.3	26.3	30.8	36.3	16.9	25.5	28.8	27.2	43.2	28.6	28.8	25.2	18.7	17.6	29.9	30.2	30.4	30.2	30.5	30.3	27.3	31.4	29.2	29.9	29.0	32.9	25.7
10.6	9.5	10.4	9.5	9.2	10.1	10.1	8.8	29.2	36.2	11.0	10.6	9.7	9.3	9.5	29.7	28.2	28.5	7.9	7.9	8.0	8.0	8.2	8.0	8.2	7.7	8.7	8.9	9.8	8.7	9.6
3.6	3.2	3.5	3.1	2.2	2.1	3.6	3.7	1.5	2.3	4.1	3.2	3.9	3.4	3.2	1.9	1.6	1.5	3.0	3.0	3.0	3.1	3.1	3.0	2.3	3.0	3.3	3.4	2.7	3.4	1.9
42.9	38.1	42.3	37.8	33.7	38.5	44.5	48.4	47.6	63.9	43.9	41.0	56.8	41.3	41.5	56.0	48.5	47.6	40.8	41.1	41.4	41.3	41.8	41.3	37.8	42.1	41.3	42.3	41.6	45.0	37.0
52.6	46.7	51.8	46.2	41.1	47.1	54.0	60.2	53.1	72.2	53.8	50.1	70.9	50.9	51.1	64.9	54.6	53.4	50.7	51.1	51.4	51.3	51.9	51.3	46.6	52.4	51.0	52.3	51.1	55.9	45.3
34.9	32.7	36.3	32.5	29.1	33.2	34.0	40.9	44.1	58.7	37.8	35.3	48.0	35.3	35.5	51.7	44.7	44.0	34.7	34.9	35.1	35.1	35.5	35.0	32.7	35.6	35.2	36.0	35.7	38.2	31.9
5.7	5.1	5.7	5.0	4.5	5.3	6.2	7.3	3.4	5.1	5.8	5.4	8.6	5.7	5.8	5.0	3.7	3.5	6.0	6.0	6.1	6.0	6.1	6.1	5.5	6.3	5.8	6.0	5.8	6.6	5.1
2.1	1.9	2.1	1.9	1.8	2.0	2.0	1.8	5.8	7.2	2.2	2.1	1.9	1.9	1.9	5.9	5.6	5.7	1.6	1.6	1.6	1.6	1.6	1.6	1.6	1.5	1.7	1.8	2.0	1.7	1.9
4	4	4	4	4	4	4	4	3	3	4	4	4	4	4	3	3	3	5	5	5	5	5	5	5	5	5	5	5	5	5
1987	1987	1987	1987	1987	1987	1987	1987	1989	1989	1985	1985	1987	1987	1987	1989	1989	1989	1987	1987	1987	1987	1987	1987	1987	1987	1987	1987	1987	1987	1987

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Table 2.4-4 (Page 1 of 5)

CAPITAL COST DISTRIBUTIONS FOR CLOSED-CYCLE GAS TURBINE

	CASE NO.	1	2	3	4	5	6	7	8	9	10
MAJOR COMPONENTS											
PRIME CYCLE											
HELIUM TURB-COMP-GEN	MM$	14.7	14.7	14.7	14.7	14.7	14.7	14.7	14.7	29.3	58.6
RECUPERATOR	MM$	15.0	15.0	15.0	15.0	15.0	15.0	15.0	15.0	30.0	60.0
PRECOOLER	MM$	2.3	2.3	2.3	2.3	2.3	2.3	2.3	2.3	4.6	9.2
BOTTOMING CYCLE											
ORGANIC OR STEAM TURB-GEN	MM$	0.	0.	0.	6.2	6.2	6.2	0.	0.	0.	0.
ORGANIC OR STEAM BOILER	MM$	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.
ORGANIC CONDENSER	MM$	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.
PRIMARY HEAT INPUT AND FUEL SYSTEM											
FURNACE MODULES	MM$	46.0	46.2	46.7	26.6	27.7	25.8	26.0	43.7	91.6	183.3
HIGH TEMP AIR PREHEATER	MM$	2.0	2.5	2.2	0.	0.	0.	0.	0.	4.0	8.1
LOW TEMP AIR PREHEATER	MM$	1.3	1.3	1.3	0.	0.	0.	0.	0.	2.6	5.2
PRESSURIZING GAS TURBINE (COMP-GEN-HEAT EXCH)	MM$	0.	0.	0.	37.5	41.8	40.2	10.7	36.8	0.	0.
GASIFIER (INCLUDING BOOST STEAM TURB-COMP	MM$	0.	0.	0.	137.4	166.7	149.6	0.	0.	0.	0.
SUB-TOTAL OF MAJOR COMPONENTS	MM$	81.3	81.9	82.2	239.7	274.3	253.8	68.7	112.5	162.2	324.3
BALANCE OF PLANT											
COOLING TOWER	MM$	2.3	2.3	2.3	2.6	2.6	2.6	2.3	2.3	4.4	8.6
ALL OTHER	MM$	36.7	37.7	36.7	42.2	46.2	36.7	19.4	46.9	69.4	132.5
SITE LABOR	MM$	13.0	13.4	13.0	16.0	17.9	12.4	7.3	16.4	24.8	47.6
SUB-TOTAL OF BALANCE OF PLANT	MM$	52.0	53.4	52.0	60.8	66.7	51.7	29.0	65.6	98.6	188.7
CONTINGENCY	MM$	26.7	27.1	26.8	60.1	68.2	61.1	19.5	35.6	52.2	102.6
ESCALATION COSTS	MM$	31.0	31.5	31.2	70.0	79.4	71.1	17.7	41.5	74.8	176.1
INTEREST DURING CONSTRUCTION	MM$	33.7	34.2	33.9	76.0	86.3	77.3	17.5	45.1	87.0	216.9
TOTAL CAPITAL COST	MM$	224.7	228.1	226.2	506.7	574.8	515.1	152.5	300.2	474.7	1008.7
MAJOR COMPONENTS COST	$/KWE	294.5	297.9	298.3	327.3	347.5	329.1	204.9	282.3	293.5	293.4
BALANCE OF PLANT	$/KWE	188.4	194.3	188.7	83.0	84.5	67.1	86.5	164.6	178.4	170.7
CONTINGENCY	$/KWE	96.6	98.4	97.4	82.1	86.4	79.2	58.3	89.4	94.4	92.8
ESCALATION COSTS	$/KWE	112.5	114.6	113.4	95.5	100.6	92.3	52.9	104.0	135.4	159.3
INTEREST DURING CONSTRUCTION	$/KWE	122.2	124.5	123.2	103.8	109.3	100.2	52.3	113.1	157.5	196.2
TOTAL CAPITAL COST	$/KWE	814.2	829.7	821.1	691.7	728.3	668.0	454.9	753.3	859.2	912.4

Table 2.4-4 (Page 2 of 5)

CAPITAL COST DISTRIBUTIONS FOR CLOSED-CYCLE GAS TURBINE

	CASE NO.	11	12	13	14	15	16	17	18	19	20
MAJOR COMPONENTS											
PRIME CYCLE											
HELIUM TURB-COMP-GEN	MM$	13.4	16.3	12.9	16.0	14.5	14.2	14.8	14.4	16.8	21.1
RECUPERATOR	MM$	20.0	15.0	37.5	15.5	14.5	13.8	14.4	15.4	12.8	11.0
PRECOOLER	MM$	2.2	2.5	2.2	2.7	2.3	2.3	2.3	1.7	1.6	2.8
BOTTOMING CYCLE											
ORGANIC OR STEAM TURB-GEN	MM$	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.
ORGANIC OR STEAM BOILER	MM$	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.
ORGANIC CONDENSER	MM$	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.
PRIMARY HEAT INPUT AND FUEL SYSTEM											
FURNACE MODULES	MM$	31.9	44.8	42.2	55.9	48.7	49.1	50.6	48.7	52.1	57.4
HIGH TEMP AIR PREHEATER	MM$	0.	0.	2.3	0.	3.5	2.3	2.0	3.6	1.9	0.
LOW TEMP AIR PREHEATER	MM$	1.5	0.	1.0	2.1	1.3	1.3	1.3	1.3	1.4	1.9
PRESSURIZING GAS TURBINE (COMP-GEN-HEAT EXCH)	MM$	0.	6.1	0.	0.	0.	0.	0.	0.	0.	0.
GASIFIER (INCLUDING BOOST STEAM TURB-COMP	MM$	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.
SUB-TOTAL OF MAJOR COMPONENTS	MM$	69.0	84.8	98.1	92.2	84.6	83.0	85.5	85.1	86.6	94.2
BALANCE OF PLANT											
COOLING TOWER	MM$	2.0	2.6	1.8	2.7	2.3	2.2	2.3	2.3	2.4	3.0
ALL OTHER	MM$	32.5	42.0	28.6	42.4	36.2	35.7	37.1	36.2	38.4	40.1
SITE LABOR	MM$	11.5	14.9	10.1	15.0	12.8	12.6	13.1	12.8	13.6	17.0
SUB-TOTAL OF BALANCE OF PLANT	MM$	46.0	59.5	40.6	60.1	51.3	50.6	52.5	51.3	54.5	68.2
CONTINGENCY	MM$	23.0	28.9	27.7	30.5	27.2	26.7	27.6	27.3	28.2	32.5
ESCALATION COSTS	MM$	26.8	26.2	32.3	35.5	31.7	31.1	32.1	31.8	32.8	37.8
INTEREST DURING CONSTRUCTION	MM$	29.1	25.9	35.1	38.1	34.4	33.8	34.9	34.5	35.7	41.1
TOTAL CAPITAL COST	MM$	193.9	225.2	233.9	256.7	229.2	225.2	232.6	230.0	237.8	273.7
MAJOR COMPONENTS COST	$/KWE	298.6	210.4	439.7	304.5	322.8	334.4	295.7	332.0	290.2	243.0
BALANCE OF PLANT	$/KWE	199.3	147.7	181.8	198.5	195.6	203.9	181.8	200.0	182.5	175.8
CONTINGENCY	$/KWE	99.6	71.6	124.3	100.6	103.7	107.7	95.5	106.4	94.5	83.8
ESCALATION COSTS	$/KWE	115.9	65.0	144.7	117.1	120.7	125.3	111.2	123.9	110.1	97.5
INTEREST DURING CONSTRUCTION	$/KWE	126.0	64.3	157.2	127.2	131.2	136.2	120.8	134.6	119.6	106.0
TOTAL CAPITAL COST	$/KWE	839.4	559.0	1047.7	847.9	874.0	907.5	805.0	896.9	796.8	706.1

Table 2.4-4 (Page 3 of 5)

CAPITAL COST DISTRIBUTIONS FOR CLOSED-CYCLE GAS TURBINE

	CASE NO.	21	22	23	24	25	26	27	28	29	30
MAJOR COMPONENTS											
PRIME CYCLE											
HELIUM TURB-COMP-GEN	MM$	25.9	14.6	14.7	17.8	14.3	11.8	14.6	12.9	14.7	16.0
RECUPERATOR	MM$	33.7	14.9	74.0	13.0	27.9	15.1	25.0	76.8	29.0	33.0
PRECOOLER	MM$	3.4	2.6	2.2	2.7	2.3	2.1	2.6	2.1	2.2	2.6
BOTTOMING CYCLE											
ORGANIC OR STEAM TURB-GEN	MM$	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.
ORGANIC OR STEAM BOILER	MM$	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.
ORGANIC CONDENSER	MM$	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.
PRIMARY HEAT INPUT AND FUEL SYSTEM											
FURNACE MODULES	MM$	57.9	47.2	48.0	46.2	44.7	30.3	33.6	44.4	49.3	52.5
HIGH TEMP AIR PREHEATER	MM$	0.	2.9	2.3	0.	0.	1.5	0.	2.5	3.0	3.3
LOW TEMP AIR PREHEATER	MM$	2.2	1.3	1.2	0.	0.	1.0	1.4	1.0	1.2	1.4
PRESSURIZING GAS TURBINE (COMP-GEN-HEAT EXCH)	MM$	0.	0.	0.	6.9	5.4	0.	0.	0.	0.	0.
GASIFIER (INCLUDING BOOST STEAM TURB-COMP	MM$	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.
SUB-TOTAL OF MAJOR COMPONENTS	MM$	123.1	83.5	142.3	86.6	94.6	61.8	77.2	139.7	99.4	108.8
BALANCE OF PLANT											
COOLING TOWER	MM$	3.5	9.2	2.1	3.0	2.3	1.7	2.2	1.8	2.2	2.5
ALL OTHER	MM$	55.8	43.2	33.2	47.1	37.1	27.9	35.8	29.3	35.0	39.2
SITE LABOR	MM$	19.8	18.0	11.8	16.7	13.2	9.9	12.7	10.4	12.4	13.9
SUB-TOTAL OF BALANCE OF PLANT	MM$	79.1	70.4	47.1	66.7	52.6	39.5	50.8	41.5	49.5	55.5
CONTINGENCY	MM$	40.4	30.8	37.9	30.7	29.5	20.3	25.6	36.2	29.8	32.9
ESCALATION COSTS	MM$	47.1	35.8	44.1	27.8	26.7	23.6	29.8	42.2	34.7	38.3
INTEREST DURING CONSTRUCTION	MM$	51.2	38.9	47.9	27.5	26.4	25.6	32.4	45.8	37.7	41.6
TOTAL CAPITAL COST	MM$	340.9	259.5	319.3	239.3	229.9	170.9	215.8	305.4	251.1	277.1
MAJOR COMPONENTS COST	$/KWE	300.8	313.2	511.7	193.1	331.7	329.7	308.3	625.8	358.8	357.6
BALANCE OF PLANT	$/KWE	193.1	264.1	169.3	148.9	184.3	210.8	202.7	185.7	178.8	182.4
CONTINGENCY	$/KWE	98.8	115.5	136.2	60.4	103.2	108.1	102.2	162.3	107.5	108.0
ESCALATION COSTS	$/KWE	115.0	134.4	158.6	62.1	93.7	125.9	119.0	189.0	125.2	125.7
INTEREST DURING CONSTRUCTION	$/KWE	125.0	146.0	172.3	61.4	92.6	136.8	129.3	205.3	136.0	136.6
TOTAL CAPITAL COST	$/KWE	832.6	973.2	1148.1	533.9	805.5	911.4	861.4	1368.1	906.3	910.4

Table 2.4-4 (Page 4 of 5)

CAPITAL COST DISTRIBUTIONS FOR CLOSED-CYCLE GAS TURBINE

	CASE NO.	31	32	33	34	35	36	37	38	39	40
MAJOR COMPONENTS											
PRIME CYCLE											
HELIUM TURB-COMP-GEN	MM$	14.3	16.3	17.8	14.7	14.7	14.7	14.7	14.7	14.7	14.7
RECUPERATOR	MM$	50.0	24.0	21.6	15.0	15.0	15.0	15.0	15.0	15.0	2.0
PRECOOLER	MM$	[illegible]	2.3	2.8	1.9	1.9	1.9	1.9	2.0	1.9	1.9
BOTTOMING CYCLE											
ORGANIC OR STEAM TURB-GEN	MM$	0.	0.	0.	2.1	2.1	2.0	2.0	1.8	2.0	3.3
ORGANIC OR STEAM BOILER	MM$	0.	0.	0.	6.0	6.0	6.0	5.0	4.0	6.0	2.0
ORGANIC CONDENSER	MM$	0.	0.	0.	7.2	7.2	7.3	6.8	6.4	7.1	9.1
PRIMARY HEAT INPUT AND FUEL SYSTEM											
FURNACE MODULES	MM$	44.8	46.0	46.1	46.0	46.0	46.0	46.0	46.0	46.0	56.7
HIGH TEMP AIR PREHEATER	MM$	0.	0	0.	2.9	2.9	2.9	2.9	2.9	2.9	0.
LOW TEMP AIR PREHEATER	MM$	0.	0.	0.	1.3	1.3	1.3	[illegible]	1.3	1.3	2.0
PRESSURIZING GAS TURBINE (COMP-GEN-HEAT EXCH)	MM$	5.0	5.9	6.7	0.	0.	0.	0.	0.	0.	0.
GASIFIER (INCLUDING BOOST STEAM TURB-COMP	MM$	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.
SUB-TOTAL OF MAJOR COMPONENTS	MM$	116.4	94.5	94.9	97.0	97.1	97.1	95.5	94.0	96.9	91.7
BALANCE OF PLANT											
COOLING TOWER	MM$	2.2	2.6	2.9	4.0	4.0	4.0	4.0	4.1	4.0	5.5
ALL OTHER	MM$	34.6	40.7	45.8	54.9	54.9	55.0	54.9	54.9	54.9	67.4
SITE LABOR	MM$	12.2	14.4	16.2	23.2	23.2	23.2	23.2	23.2	23.2	28.6
SUB-TOTAL OF BALANCE OF PLANT	MM$	49.0	57.7	64.9	82.1	82.1	82.2	87.1	82.2	82.1	101.5
CONTINGENCY	MM$	33.1	30.4	32.0	35.8	35.8	35.9	35.5	35.2	35.8	38.6
ESCALATION COSTS	MM$	30.0	27.6	29.0	51.4	51.4	51.4	51.0	50.5	51.3	55.4
INTEREST DURING CONSTRUCTION	MM$	29.7	27.3	28.7	59.8	59.8	59.8	59.3	58.8	59.7	64.5
TOTAL CAPITAL COST	MM$	258.1	237.6	249.5	326.2	326.2	326.3	323.4	320.8	325.9	351.7
MAJOR COMPONENTS COST	$/KWE	359.5	235.3	212.4	281.8	283.9	285.8	282.3	282.8	284.6	224.9
BALANCE OF PLANT	$/KWE	151.3	143.7	145.3	238.5	240.2	241.8	242.6	247.1	241.3	249.0
CONTINGENCY	$/KWE	102.2	75.8	71.5	104.1	104.8	105.5	105.0	106.0	105.2	94.8
ESCALATION COSTS	$/KWE	92.7	68.8	64.9	149.2	150.3	151.3	150.6	152.0	150.8	135.9
INTEREST DURING CONSTRUCTION	$/KWE	91.7	68.0	64.2	173.6	174.9	176.0	175.1	176.8	175.5	158.1
TOTAL CAPITAL COST	$/KWE	797.4	591.6	558.3	947.2	954.0	960.5	955.6	964.6	957.4	862.7

Table 2.4-4 (Page 5 of 5)

CAPITAL COST DISTRIBUTIONS FOR CLOSED-CYCLE GAS TURBINE

	CASE NO.	41	42	43	44	45	46
MAJOR COMPONENTS							
PRIME CYCLE							
HELIUM TURB-COMP-GEN	MM$	14.7	14.7	14.7	14.7	14.7	14.7
RECUPERATOR	MM$	29.0	15.0	15.0	2.0	28.8	0.
PRECOOLER	MM$	1.9	2.6	2.6	2.6	2.6	2.6
BOTTOMING CYCLE							
ORGANIC OR STEAM TURB-GEN	MM$	1.9	3.8	2.5	5.0	2.5	14.4
ORGANIC OR STEAM BOILER	MM$	4.0	3.5	3.5	4.1	3.6	10.7
ORGANIC CONDENSER	MM$	0.	0.	0.	0.	0.	0.
PRIMARY HEAT INPUT AND FUEL SYSTEM							
FURNACE MODULES	MM$	49.3	46.0	46.0	56.7	49.3	63.7
HIGH TEMP AIR PREHEATER	MM$	3.0	2.9	2.9	0.	3.0	0.
LOW TEMP AIR PREHEATER	MM$	1.2	1.3	1.3	2.0	1.2	2.4
PRESSURIZING GAS TURBINE (COMP-GEN-HEAT EXCH)	MM$	0.	0.	0.	0.	0.	0.
GASIFIER (INCLUDING BOOST STEAM TURB-COMP	MM$	0.	0.	0.	0.	0.	0.
SUB-TOTAL OF MAJOR COMPONENTS	MM$	105.0	89.8	88.5	87.1	105.7	108.4
BALANCE OF PLANT							
COOLING TOWER	MM$	4.4	2.9	4.6	3.7	2.5	8.1
ALL OTHER	MM$	52.2	47.6	47.3	58.7	45.4	86.8
SITE LABOR	MM$	22.1	17.8	17.8	21.9	17.0	32.6
SUB-TOTAL OF BALANCE OF PLANT	MM$	78.7	68.3	69.8	84.4	64.9	127.4
CONTINGENCY	MM$	36.7	31.6	31.6	34.3	34.1	47.2
ESCALATION COSTS	MM$	52.7	45.4	45.4	49.2	48.9	67.6
INTEREST DURIN 'ONSTRUCTION	MM$	61.3	52.8	52.8	57.2	56.9	78.7
TOTAL CAPITAL COST	MM$	334.4	287.9	288.1	312.1	310.5	429.3
MAJOR COMPONENTS COST	$/KWE	311.7	288.3	290.8	256.1	354.5	205.4
BALANCE OF PLANT	$/KWE	233.8	219.4	229.2	248.2	217.5	241.4
CONTINGENCY	$/KWE	109.1	101.5	104.0	100.9	114.4	89.4
ESCALATION COSTS	$/KWE	156.5	145.6	149.1	144.6	164.1	128.2
INTEREST DURING CONSTRUCTION	$/KWE	182.0	169.4	173.5	168.3	190.9	149.1
TOTAL CAPITAL COST	$/KWE	993.1	924.2	946.7	918.1	1041.3	813.5

Table 2.4-5 (Page 1 of 2)

POWER OUTPUT AND AUXILIARY POWER DEMAND FOR BASE CASE AND PARAMETRIC VARIATIONS: CLOSED-CYCLE GAS TURBINE

CASE NO.		1	2	3	4	5	6	7	8	9	10
PRIME CYCLE POWER OUTPUT	MW	300.0	300.0	300.0	300.0	300.0	300.0	300.0	300.0	600.0	1200.0
BOTTOMING CYCLE POWER OUTPUT	MW	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.
FURNACE POWER OUTPUT	MW	0.	0.	0.	441.1	498.0	479.8	41.1	110.7	0.	0.
BALANCE OF PLANT AUX. POWER REQ'D.	MW	4.4	4.4	4.4	4.9	4.8	4.8	4.1	4.4	8.2	15.9
FURNACE AUX. POWER REQ'D.	MW	18.2	19.1	18.6	0.	0.	0.	0.	5.8	36.3	72.6
TRANSFORMER LOSSES	MW	1.5	1.5	1.5	3.7	4.0	3.9	1.8	2.1	3.0	6.0
NET STATION OUTPUT	MW	276.0	275.0	275.5	732.5	789.2	771.1	335.2	398.5	552.5	1105.5

CASE NO.		11	12	13	14	15	16	17	18	19	20
PRIME CYCLE POWER OUTPUT	MW	252.5	362.4	242.8	329.9	285.9	271.6	31[illegible].2	280.2	323.4	418.4
BOTTOMING CYCLE POWER OUTPUT	MW	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.
FURNACE POWER OUTPUT	MW	0.	46.9	0.	0.	0.	0.	0.	0.	0.	0.
BALANCE OF PLANT AUX. POWER REQ'D.	MW	4.2	4.3	4.2	4.5	4.4	4.4	4.4	4.4	4.4	4.9
FURNACE AUX. POWER REQ'D.	MW	16.1	0.	14.2	21.0	17.9	17.7	18.3	18.0	19.0	23.8
TRANSFORMER LOSSES	MW	1.3	2.1	1.2	1.6	1.4	1.4	1.6	1.4	1.6	2.1
NET STATION OUTPUT	MW	231.0	402.9	223.2	302.7	262.2	248.2	289.0	256.4	298.4	387.6

CASE NO.		21	22	23	24	25	26	27	28	29	30
PRIME CYCLE POWER OUTPUT	MW	444.6	294.4	300.0	402.6	290.9	206.3	274.1	242.8	300.0	329.9
BOTTOMING CYCLE POWER OUTPUT	MW	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.
FURNACE POWER OUTPUT	MW	0.	0.	0.	52.5	0.	0.	0.	0.	0.	0.
BALANCE OF PLANT AUX. POWER REQ'D.	MW	5.4	8.2	4.0	4.4	4.1	4.0	4.5	3.8	4.2	4.5
FURNACE AUX. POWER REQ'D.	MW	27.6	18.0	16.4	0.	0.	13.8	17.7	14.5	17.3	19.4
TRANSFORMER LOSSES	MW	2.2	1.5	1.5	2.4	1.5	1.0	1.4	1.2	1.5	1.6
NET STATION OUTPUT	MW	409.4	266.7	278.1	448.3	285.3	187.5	250.5	223.3	277.0	304.3

Table 2.4-5 (Page 2 of 2)

POWER OUTPUT AND AUXILIARY POWER DEMAND
FOR BASE CASE AND PARAMETRIC VARIATIONS:
CLOSED-CYCLE GAS TURBINE

CASE NO.		31	32	33	34	35	36	37	38	39	40
PRIME CYCLE POWER OUTPUT	MW	290.9	362.4	402.6	300.0	300.0	300.0	300.0	300.0	300.0	300.0
BOTTOMING CYCLE POWER OUTPUT	MW	0.	0.	0.	68.1	65.7	63.5	62.2	56.4	64.1	136.8
FURNACE POWER OUTPUT	MW	38.5	45.5	51.1	0.	0.	0.	0.	0.	0.	0.
BALANCE OF PLANT AUX. POWER REQ'D.	MW	3.9	4.1	4.4	3.7	3.7	3.7	3.7	3.7	3.7	4.6
FURNACE AUX. POWER REQ'D.	MW	0.	0.	0.	18.2	18.2	18.2	18.2	18.2	18.2	22.4
TRANSFORMER LOSSES	MW	1.7	2.1	2.4	1.8	1.8	1.8	1.8	1.9	1.8	2.2
NET STATION OUTPUT	MW	323.8	401.7	446.9	344.3	341.9	339.8	338.5	332.6	340.4	407.7

CASE NO.		41	42	43	44	45	46
PRIME CYCLE POWER OUTPUT	MW	300.0	300.0	300.0	300.0	300.0	300.0
BOTTOMING CYCLE POWER OUTPUT	MW	59.5	35.5	28.9	68.6	21.0	268.0
FURNACE POWER OUTPUT	MW	0.	0.	0.	0.	0.	0.
BALANCE OF PLANT AUX. POWER REQ'D.	MW	3.6	4.1	4.7	4.4	3.9	4.1
FURNACE AUX. POWER REQ'D.	MW	17.3	18.2	18.2	22.4	17.3	33.3
TRANSFORMER LOSSES	MW	1.9	1.7	1.7	1.8	1.6	2.8
NET STATION OUTPUT	MW	336.7	311.5	304.3	339.9	298.2	527.8

construction time. Both organic fluid and steam bottoming cycles were examined in parametric Cases 34 through 46.

Rationale for Point Variations

The assignment of a fixed helium compressor inlet flow of 1031 lb/s (467.6 kg/s) resulted in a wide variation of plant output with other major variables as shown in Table 2.4-6. The discussion of results to follow shows the several categories of patterns within these selections.

Table 2.4-6

CLOSED-CYCLE GAS TURBINE RANGE OF PARAMETRIC VARIATIONS

System Parameters	Base Case	Variations
Plant Variables		
Power output (MW)	276	187 - 1105
Application	Base Load	—
Coal types	Illinois #6	N. Dakota, Montana
Coal Conversion		
Direct combustion:	Atmospheric fluidized bed	Pressurized fluidized bed
Pressure-fired gas:	—	High Btu Low Btu—Integrated
Helium Cycle		
Compressor pressure ratio	2.5	2,3
Intercooled	—	2.5,4,6.25
Compressor inlet temperature (°F)	50	60,88,110
Recuperator effectiveness	85%	0%,60%,90%,95%
Recuperator pressure loss	3%	5%,7%
Bottoming Cycle		
Organic fluid temperature (°F)	—	390 - 460
Steam temperature (°F)	—	384 - 900
Heat Rejection		
Type cooling tower	Wet	Dry

DISCUSSION OF RESULTS

Overview

Table 2.4-7 presents a variety of cases that were most economically attractive. It was notable that neither the 1350 F (1005 K) nor the 1700 F (1200 K) alternative proved more economic than the base value of 1500 F (1189 K) for the helium turbine inlet. The Case 40 with organic bottoming had the highest efficiency but also the highest capital cost per kilowatt. With the exception of that one case the most economic overall efficiencies were all close to 31 percent. The greatest economy was realized with Case 20, which had a pressure ratio of 4 and an intercooled compressor cycle. The pressurized-furnace low-Btu Case 4 had a combustion gas turbine and a steam turbine producing power as well as the helium gas turbine in the integrated plant. In fact, of the 732 MW net station power output only 300 MW was produced by the helium gas turbine. The most economic bottoming cycle for the helium gas turbine was a steam cycle that completely eliminated the helium recuperator.

Table 2.4-7

OVERVIEW OF MOST ECONOMIC CASES

Case	Configuration*	Production Cost (mills/kWh)	Plant Cost ($/kW)	Plant Output (MW)	Thermo-dynamic Effi-ciency (%)	Over-all Effi-ciency (%)
1	Atmospheric fluidized bed base case	38.8	814	276	36.4	29.5
40	Organic bottoming, 60% recuperator	37.8	862	408	42.6	35.3
46	Steam bottoming, no recuperator	37.0	813	528	37.6	30.8
8	Pressurized fluidized bed, recuperative	35.9	753	398	36.4	31.8
4	Pressurized furnace, LBtu, integrated	34.5	691	732	36.4	31.0
20	Intercooled compressor, 4 pressure ratio	33.7	706	388	38.3	31.6

*All at 1500 F, 2.5 pressure ratio, Illinois #6 coal, atmospheric fluidized bed except as noted

Cycles with Atmospheric Fluidized Beds. Table 2.4-8 presents the economic and overall thermal performance for nine cases using atmospheric fluidized bed heat sources. The intercooled high pressure ratio compressor Case 20 was most economic.

Table 2.4-8

RESULTS FOR CLOSED-CYCLE GAS TURBINES WITH ATMOSPHERIC FLUIDIZED BED HEAT SOURCES

Pressure Ratio	Recuperator Efficiency 85% Turbine Temperature 1350 F (mills/kWh, overall efficiency, case)	Recuperator Efficiency 85% Turbine Temperature 1500 F (mills/kWh, overall efficiency, case)	Recuperator Efficiency 90% Turbine Temperature 1500 F (mills/kWh, overall efficiency, case)
2	43.9,26.4%,No.26	46.5,30.6%,No.13	56.8,29.9%,No.28
2.5	40.2,27.9%,No.11	38.8,29.5%,No.1	41.3,31.1%,No.29
3	41.0,27.4%,No.27	40.2,28.0%,No.14	41.5,30.5%,No.30
4, Intercooled Compressor		33.7,31.6%,No.20	—

Pressure-Fired High-Btu Gas Cases. None of these cases as presented in Table 2.4-9 merited inclusion in the most economic cases of Table 2.4-7. The pressurizing combustion gas turbine set had a pressure ratio of 10 and a turbine inlet temperature of 1200 F (922 K). The power plant efficiency was comparable to other configurations; the high price of high-Btu gas fuel resulted in the high cost for electricity production.

Table 2.4-9

CLOSED-CYCLE GAS TURBINES WITH PRESSURE FIRED HIGH-BTU HEAT SOURCES

Pressure Ratio	Recuperator Efficiency 85% Turbine Temperature 1500 F (mills/kWh, overall efficiency, case)	Recuperator Efficiency 85% Turbine Temperature 1700 F (mills/kWh, overall efficiency, case)	Recuperator Efficiency 90% Turbine Temperature 1700 F (mills/kWh, overall efficiency, case)
2	—	63.9,24.5%,No.25	56.8,29.9%,No.31
2.5	46.7,29.2%,No.7	48.3,30.6%,No.12	48.5,31.5%,No.32
3	—	47.6,30.4%,No.24	47.6,31.1%,No.33

Sensitivity to Variables. Many cases were evaluated to determine the sensitivity to variations from the base case. Table

2.4-10 presents the differentials from the 38.8 mills/kWh and 276 MW of the base case at 1500 F (1189 K), 2.5 pressure ratio with atmospheric fluidized bed heat source.

Table 2.4-10

SENSITIVITY TO VARIATIONS FROM THE BASE CASE

Variations from Base Case	Effect of Variations (mills/kWh)
1350 F	Adds 1.4 over 1500 F
Pressurized furnace, HBtu	Adds 7.9 over AFB
Pressurized furnace, LBtu	Reduces 4.3 over AFB
Pressurized fluidized bed with recuperator	Reduces 2.9 over AFB
Dry cooling tower	Adds 5.7 over wet cooling tower
Recuperator 90% effectiveness	Adds 2.5 over 85% effective recuperator
Recuperator 5% pressure loss	Adds 2.5 over 3 percent pressure loss recuperator
Organic bottoming	Reduces 1.0 over base; adds 132 MW over base
Steam bottoming	Reduces 1.8 over base; adds 252 MW over base

RECOMMENDED CASES

Case 20, featuring an intercooled compressor with a pressure ratio of 4, firing Illinois No. 6 coal in an atmospheric fluidized bed, was recommended. The capital cost of 706 dollars per kilowatt and production cost of 33.7 mills per kilowatthour were lowest for helium closed-cycle gas turbines with 1500 F (1189 K) turbine inlet.

2.5 SUPERCRITICAL CO_2 CYCLE

DESCRIPTION OF CYCLE

The supercritical CO_2 cycle is a variation of the closed gas turbine concept previously described. Its mode of operation is similar to that of other closed-cycle gas turbines, so its cycle components and arrangement diagrams are similar to those for the helium closed-cycle gas turbine. The differentiation is that the supercritical CO_2 cycle is intended to operate very close to the critical point to achieve a reduction in the compressor power requirement. Both cycles have the following common attributes:

- Exhausting heat at high temperatures that enhance use of dry cooling towers
- Requiring thermal regenerators to achieve acceptable efficiencies
- Having sensitivity to pressure losses anywhere in the system (although not so sensitive as the helium gas turbine)
- Requiring high turbine inlet temperatures (1400 F [1033 K] to 1500 F [1089 K]) as compared with closed cycles in present-day utility service.

CO_2 is most attractive working fluid because of its relatively low critical pressure (1070 psia [7.38 MN/m^2]), its availability, and its heat transfer and thermodynamic attributes. The pressure level at all points in this cycle is above the critical pressure, thus precluding condensation.

The regenerator exchanges heat between the high-temperature fluid exhausting from the turbine and the high-pressure fluid en route to the primary heater. The heat transferred in the regenerator is approximately twice that of the primary heater, and its effective temperature difference for heat transfer must be low in order to obtain high heater exchange effectiveness. The regenerator is a critical cycle component. A high effectiveness must be achieved with a minimal pressure drop, while being designed to withstand the high system pressures. In addition, turbine trip-outs due to the loss of load would impose abrupt pressure changes and temperature changes on the large heat exchange components.

The supercritical CO_2 gas turbine cycle has the advantage over the other closed gas turbine cycles of having a fluid density entering the compressor that is quite high, about three-quarters that of water. Thus the work of compression is only 20 percent of the total turbine output work, as compared with approximately 50 percent in other closed gas turbine cycles. This makes the cycle less sensitive to compressor efficiency.

Although the rotational equipment in this cycle will be substantially different from the pumps and turbines in a normal steam cycle and will require major design efforts, the key element for cycle success will be the heat exchange equipment design.

The primary heat input heat exchanger is the most critical heat exchanger in the cycle. The high-temperature and high-pressure operation will require the use of superalloy materials such as Hastelloy X for the tube material. These materials are very difficult to work with, and problems such as a requirement for heat treatment after a welding operation would make fabrication very costly.

The regenerator operates at lower temperature but has twice the heat exchange capability of the primary heat exchanger. A shell-in-tube design was assumed appropriate to withstand the high differential pressure resulting from 3800 psi (26.2 MN/m^2) fluid on the tube side and 1300 psi (8.96 MN/m^2) on the shell side.

The heat rejection for this cycle occurs at temperatures that suggest direct air cooling. As compared with condensing cycles of any kind, the noncondensing cycle would only require approximately one-half to one-fourth of the surface for dry coolers and the opportunity might exist to operate with natural draft in place of the more conventional forced draft dry cooling towers.

Other cycle configurations are possible for supercritical, noncondensing power systems. A reduction of primary heater pressure could be realized by expanding the working fluid through a turbine placed between the regenerator and the primary heater. The primary heater would restore the energy extracted in this turbine by heating the fluid. The original turbine would thus have a reduced pressure ratio. In this system, only the regenerator, at modest temperature levels, would be exposed to the extreme fluid pressure.

The characteristic of sensible heat rejection from the cycle establishes a good match with a dry cooling tower. However, unlike the standard Brayton cycle, the low temperature exiting the "pump" permits regeneration of the low-pressure fluid to a temperature below the point where effective coupling with bottoming cycles can be achieved. Therefore no bottoming cycles were considered in this system.

Rationale for Point Variations

The cycle schematic which was employed for the base-case evaluation is shown in Figure 2.5-1. The configuration features a recompression cycle. In this concept, the low-pressure flow is split as it exits the low-temperature recuperator. A portion of this flow goes directly to a compressor; the remaining portion continues to a heat rejection system before being raised in pressure

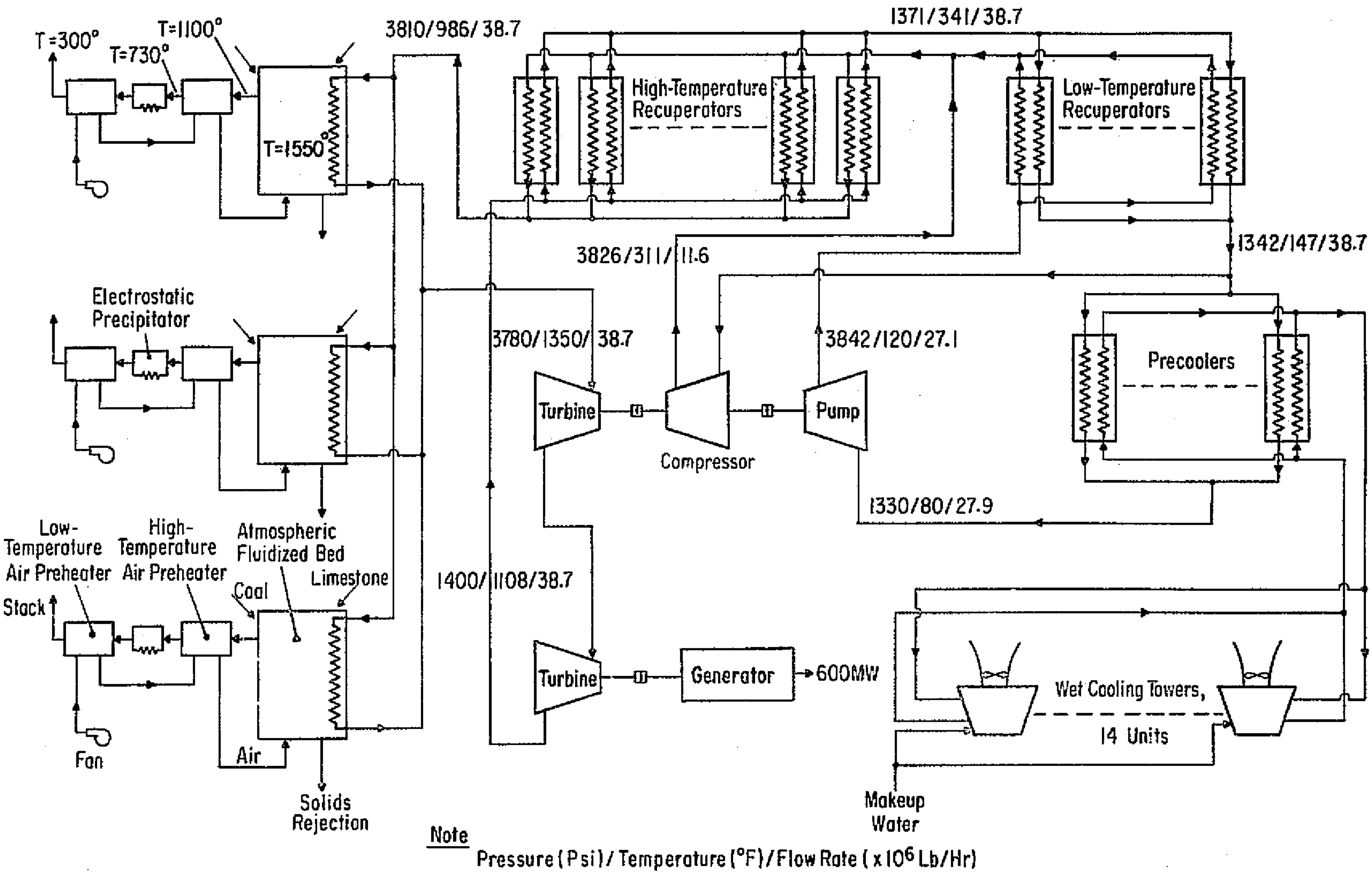

Figure 2.5-1. Supercritical CO_2 Cycle

in a pump. This configuration provides for a flow mismatch in the low-temperature recuperator and results in the acheivement of more effective regeneration and subsequently higher efficiencies with a slight reduction in specific power output. The base-case conditions were: generator output 600 MW, turbine inlet conditions 1350 F (1005 K) and 3780 psi (26.1 MN/m^2), turbine outlet pressure 1400 psi (9.67 MN/m^2). Parametric variations were selected from this base case to evaluate cycle variations.

The atmospheric fluidized bed with direct coal combustion was the base-case furnace for investigations of the supercritical CO_2 cycle. The use of clean fuels was explored by employing a pressurized furnace with both high- and low-Btu fuel. The pressurized fluidized bed was also explored in a regenerative mode as another direct coal-burning cycle.

The temperature span of 1200 F to 1600 F (922 K to 1144 K) matches the lowest temperature at which this cycle is thermodynamically competitive. The upper limit approaches the point where working fluid disassociation problems could begin.

A wide range was given to both the $\Delta P/P$ total and the $\Delta P/P$ in the recuperator. This was done in order to develop a background of information to permit trade-off studies in the heat exchanger components due to economic considerations and definition of cycle optimum.

A variation in pump flow fraction was considered. The flow fraction limit of 1.0 eliminates the auxiliary compressor and permits evaluation of the basic supercritical CO_2 cycle.

The use of a post heat configuration implies that the flow exiting the high-temperature recuperator is partially expanded in a pump and compressor drive turbine before being introduced into the primary heat exchanger at a lower pressure for energy addition and subsequent expansion in the generator drive turbine. This concept permits a significant reduction of stress levels in the primary heat exchanger due to the reduced pressure at the expense of cycle efficiency.

A cycle configuration was evaluated for use with a dry cooling tower as well as the base-case wet cooling tower. With the base-case design, the pump inlet temperature was varied to show the off-design performance at varying ambient conditions.

The only secondary cycle which was utilized with the supercritical CO_2 cycle was a steam plant which operates on the exhaust energy from the gas turbine of the pressurized furnace. This combined gas-turbine/steam-turbine case is similar to the combined cycle configuration.

ANALYTICAL PROCEDURES AND ASSUMPTIONS

The performance of a supercritical CO_2 cycle is obtained by making appropriate modifications to the ideal thermodynamic cycle

to permit inclusion of component efficiencies and parasitic losses. The nonreversible processes which are introduced into the calculation include:

- Turboequipment inefficiency
- Seal leakage
- Realistic approach temperatures in heat exchangers
- Pressure drops in flow lines
- Mechanical and bearing losses

With input values for these "real" component effects, turbine pressure ratio and inlet temperature, and pump inlet temperature, the thermodynamic cycle characteristics are calculated. The thermal transport of all of the heat exchanger equipment is determined in addition to the specific power output and thermal efficiency. An assumed energy output from the cycle then sets the flow rate and allows sizing of the individual components.

The calculations are based upon updated properties for supercritical CO_2 which are available in tabular and computerized format. This information has been compiled from the best available information and represents a critical data base for these evaluations. The property data are utilized in all analyses.

The initial evaluations for turboequipment efficiency followed the procedure employed by Actron, Inc. This approach utilizes the analytical procedures outlined by Baljé (ref. 1). A constant per stage pressure ratio is assumed and the design is set to achieve a per stage specific speed of 50 to 170. The specific diameter is then chosen to permit a 90 percent stage efficiency. The number of stages is dictated by blade bending loading constraints and not by aerodynamic considerations (due to low flow Mach numbers). The pump design considerations have been verified on a preliminary basis from initial Actron, Inc., experiments, and the calculated pump performance will be reviewed and utilized.

In the Task I Study, the efficiencies of the rotational components were assumed and treated as input variables.

The thermodynamic evaluations of the cycle set the "four corner" property conditions on the heat exchangers. The configuration selected for this study was a shell and tube design. The calculation procedure was to segment the heat exchanger and perform a stepwise calculation through the heat exchanger utilizing a log mean temperature difference approach. This segmented model permitted proper assessment of the changing property values throughout the heat exchanger. The heat transfer coefficient was calculated by a standard equation for forced convection heat transfer correlation. This approach has been verified by open literature studies (refs. 2, 3). The pressure drop was similarly handled by a standard fluid flow approach. An iterative procedure was employed until the corner points were matched.

A major design criterion for tube size was the stress levels at the high temperatures and pressures in the heat input and recuperative heat exchangers. The heat input heat exchanger and fuel combustion process were designed and evaluated by the Foster Wheeler Energy Corporation. Therefore, this component was evaluated on a common basis with all of the closed cycles. The initial design of the recuperator was established by the Advocate Team. The mechanical design portion of their heat exchanger program contained a data bank of material properties. The tube design stress was set by the advocates' interpretation of the ASME code, Section 8.

The final design and costing of the recuperators and precoolers were performed by the Heat Transfer Products Operation of the General Electric Company.

The parameters and assumptions employed for the base case are presented in Table 2.5-1.

Table 2.5-1

PARAMETERS FOR BASE CASE

Pump Inlet Temperature—80 F

Turbine Inlet Temperature—1350 F

Pump Inlet Pressure—1330 psia

Pump Discharge Pressure—3842 psia

Pressure Losses (total)—120 psi

Recuperator Minimum Stream Temperature Difference—20 F

Internal Turbine Leakage as a Percentage of the Turbine Flow—2%

Generator Efficiency—98%

Power Turbine Mechanical Efficiency—98%

Power Turbine Efficiency—90%

Pump Drive Turbine Efficiency—90%

Pump Efficiency—90%

Compressor Efficiency—87%

Overall Conversion System Thermal Losses as a Percentage of Net Electrical Output—0.1%

DESIGN AND COST BASIS

Turboequipment

The high initial and exhaust pressure of the CO_2 turbine results in a compact turbine design with relatively few, but extremely heavily loaded, stages. The small energy range and high

working fluid density result in a small radius ratio (small radial height of blade to pitch diameter) and low Mach number stages. However, to achieve the desired turbine efficiency, reasonable bucket aspect ratios must be used, and these ratios result in bucket gas bending stresses many times higher than those used in current steam turbine practice.

Since no firm design basis existed for this equipment, the design cost basis was extrapolated from steam turbine practice.

Power Drive Turbine. This unit is employed for generator drive. It was intended to operate at 3600 RPM achieving an assumed efficiency of 90 percent. The configuration which was selected was a noncooled, double-flow design.

The pressure and temperature conditions on this turbine very closely match those of the high-pressure turbine for the advanced steam cycle as a reference design. The significantly higher volume flow for the supercritical CO_2 turbine requires the double-flow arrangement. Since this turbine is direct coupled to the generator, interrupt valves will be required in the flow line upstream of the turbine to prevent turbine overspeed during loss of load accidents.

The cost projections were a strong function of inlet turbine temperature, the increment being \$0.37/kW/°F (\$0.21/kW/°K). The cost base was \$216/kWe at 1300 F (977 K) turbine inlet temperature.

Pump Drive Turbine. This unit could probably be configured in a single-stage double-flow arrangement. The design was again based upon a noncooled configuration. The same reference turbine was employed as in the power drive turbine. In this particular turbine, a compressor and pump bypass system might eliminate the requirement for high-temperature valving for inlet flow interruption. With the elimination of the high-temperature control system, the reference design for this unit was \$185/kWm at 1350 F (1005 K).

Compressor and Pump. Both of these units are operated at low temperature and modest pressure ratio; they are therefore assumed to be relatively simple units. Their cost was assumed to be approximately \$10/kWm.

The compressor could be a two-stage design with an assumed efficiency of 87 percent. The pump would be a single-stage design with an assumed efficiency of 90 percent.

Heat Exchangers

Recuperators. The recuperators were divided into low- and high-temperature units. Because of the high temperatures and pressures, combined with the large amounts of thermal energy transfer, the units were significant cost items.

A shell and tube heat exchanger configuration was utilized with a cross-counter flow arrangement. The unit was designed for 100 psi (0.685×10^6 N/m^2) over the operating pressure. This configuration was required for the high working pressure levels.

A series of parallel units were employed. The number of units was set by a stress limitation in the tube sheet. And the tube sheet was limited to 14-in. (0.356 m) thickness from fabrication considerations. The shell diameter was 32 in. (0.813 m).

The initial design performed by the advocate was modified to include allowances for additional shell side pressure drops. This was required to include allowances for flow distribution and tube supports. A fouling factor of 0.0003 F ft^2 hr/Btu (5.29×10^{-5} m^2 k/watt was also assigned to both the tube and shell side.

The basic configuration of the shell was assumed to be in a U-tube configuration. The tube length was 50 ft (15.2 m) and the tube diameter was 1/2 in. (0.013 m).

The heat exchanger cost was a strong function of temperature. Table 2.5-2 gives the temperature, material, and cost relationships.

Table 2.5-2

HEAT EXCHANGER MATERIALS

Tube Temperature		Approximate Fabricated Heat Exchange Surface Costs (\$/ft^2)
$T_t < 800$ F	Carbon Steel	30
$800 < T_t < 1100$	Stainless Steel	200
$T_t > 1100$	Inconel	300

In order to reduce the cost of the high-temperature recuperator, a series arrangement was employed. This permits a high-temperature unit to be designed at approximately \$200/ft^2* (\$2153) m^2) and a lower temperature unit to be designed at approximately \$30/ft^2 (\$323/m^2). In addition, 80 percent of the allowable shell side pressure drop was assigned to the higher temperature unit. These modifications resulted in an average heat exchanger cost of approximately \$115/ft^2 (\$1238/m^2).

Precoolers. The precooler design was a water-to-CO_2 heat exchanger with the water on the shell side of the shell and tube unit. This heat exchange was accomplished in modular units. The low-temperature range for heat rejection will permit the utilization of 90/10 copper nickel at a fabrication cost of \$30/ft^2 (\$323/m^2).

*Square foot of heat transfer area.

Furnace—Primary Heat Input Heat Exchanger. As with all closed-cycle concepts this unit was designed and costed by Foster Wheeler Energy Corporation. The base case employed an atmospheric fluidized bed with heat exchange occurring both within the bed and in the convective space above the bed.

As a result of the high temperature of the working fluid entering the furnace, the combustion gas temperature exiting the furnace was approximately 1100 F (866.5 K). A high-temperature air preheater was therefore required prior to entering the electrostatic precipitator. This heat exchange unit was a tubular construction and was estimated to cost \$2.5 x 10^6. The modular cost for the AFB unit for the base case was \$18.9 x 10^6.

RESULTS

The analyses of the advocate, furnace designer, and architect-engineer were combined to provide the system performance and economics of the supercritical CO_2 cycle. Table 2.5-3 gives the summary of results for the base case. In addition to the performance and cost and major component characteristics, this figure gives values for natural resource required and environmental entrusion. The emissions and wastes from this cycle are from the atmospheric fluidized bed and are within the allowable limits.

The results for the thirty-two parametric variations are shown in Table 2.5-4. The capital cost distribution for these points are shown in Table 2.5-5.

DISCUSSION OF RESULTS

The supercritical CO_2 cycle achieved a good overall efficiency. The overall (coal pile to bus bar) efficiencies were in the 40 to 42 percent range for the cases investigated. Thermodynamic cycle efficiency of 48 to 50 percent was reduced to the 40 percent level when power plant losses were accounted for, e.g., furnace stack losses, and furnace and balance of plant auxiliary electrical requirements. Table 2.5-6 presents the power output and auxiliary demands for the base case and parametric variations. This efficiency level was approximately 5 percentage points better than a conventional steam power plant designed to meet the environmental constraints.

The high density working fluid entering the pump resulted in low regenerative mechanical work to perform the pumping operation. This amounts to only 20 percent of potential turbine output. This is higher than would be expected for a liquid (Rankine Cycle) system but less than half that of a closed gas turbine (Brayton Cycle).

The thermal regeneration was however very high. Approximately 2.6 times the thermal input had to be regenerated in the recuperators. This thermal transport occurs at high pressures and temperatures thus making the design of the heat exchangers more complex.

Table 2.5-3

SUMMARY SHEET
SUPERCRITICAL CO_2 CYCLE BASE CASE

CYCLE PARAMETER

Net Power Output (MWe)	566
Furnace and Coal Type	Atmospheric fluidized bed Illinois No. 6
Prime Cycle	
Turbine inlet temperature (°F)	1350
Compressor pressure ratio	2.7
Recuperator pressure drop ($\Delta p/p$)	0.05
Primary heat exchanger Δp (psi)	30
Precooler Δp (psi)	12
Pump flow fraction	0.7
Compressor inlet temperature (°F)	80
Recuperator minimum temperature difference (°F)	20
Turbine efficiency	0 9
Heat Rejection	Wet cooling tower

MAJOR COMPONENT CHARACTERISTICS

Major Component	Unit or Module: Size (ft) (W x L (or D) x H)	Weight (lb) (x 10^6)	Cost ($ x 10^6)	Units Required	Total Cost ($ x 10^6)	$/kW Output
Prime Cycle						
CO_2 turbine-generator	60 x 15	1.6	130.0	1.0	130.0	228
CO_2 turbine drive-pump-compressor			20.6	1.0	20.6	36
High-temperature recuperator	14 x 34 x 4	0.17	1.0	160.0	166.4	294
Low-temperature recuperator	23 x 42 x 7	0.30	1.1	32.0	36.0	63
Primary Heat Input System						
Furnace module	12 x 30 x 140	3.5	18.89	2.8	53.0	93
High-temperature air preheater	30 x 58 x 7	0.26	0.36	19.6	7.1	12

PERFORMANCE AND COST

Thermodynamic efficiency (percent)	48
Powerplant efficiency (percent)	40
Overall energy efficiency (percent)	40
Plant capital cost ($ x 10^6)	1072
Plant capital cost ($/kWe)	1894
Cost of electricity (mills/kWh)	69.3

NATURAL RESOURCES

Coal (lb/kWh)	0.79
Water (gal/kWh)	
Total	0.44
Cooling	0.44
Processing	
Makeup	
NO_x suppression	
Stack gas cleanup	
Land (acres/100 MWe)	7.0

ENVIRONMENTAL INTRUSION

	Lb/10^6-Btu Input	Lb/kWh Output
SO_2	1.03	8.45×10^{-3}
NO_x	0.27	2.26×10^{-3}
HC	·	--
CO	0.21	1.77×10^{-3}
Particulates	0.1	0.85×10^{-3}

	Btu/kWh
Heat to water	3937
Heat, total rejected	5118

Wastes	Lb/kWh	Lb/Day
Furnace solids	0.143	1.95×10^6
Fine dust from cyclones	0.108	1.46×10^6
Fly ash	0.015	0.02×10^6

FOLDOUT FRAME

Parameters	Case 1*	2	3	4	5	6	7
Power Output (MWe)	566	1132	565	563	750	1261	1320
Furnace, Coal, and Conversion Process	AFB Ill. #6	→	AFB Mont	AFB N.D.	$(PFB)_R$ Ill. #6	PF Ill. #6 LBtu	PF Mont LBtu
Prime Cycle							
Turbine inlet temperature (°F)	1350						
Compressor pressure ratio	2.7						
Recuperator pressure drop ($\Delta p/p$)	0.05						
Primary heat exchanger (Δp psi)	20						
Precooler (Δp psi)	12						
Pump flow fraction	0.7						
Compressor inlet temperature (for performance variations) (°F)	80						
Change in heat input heat exchanger design	--	--	--	--	--	--	--
Heat rejection	WCT						
Recuperator minimum temperature difference (°F)	20						
Turbine efficiency	0.9						
PF Air Supply							
Excess air (percent)	--	--	--	--	20	15	
Pressure ratio	--	--	--	--	10		
Turbine inlet temperature (°F)	--	--	--	--	1600	1800	
Regenerator efficiency	--	--	--	--	0.85	Steam	
Actual Powerplant Output (MWe)	566	1132	565	563	750	1261	1320
Thermodynamic Efficiency (percent)	47.7	47.7	47.7	47.7	47.7	47.7	47.7
Powerplant Efficiency (percent)	40.0	40.0	38.5	35.7	39.2	35.0	35.3
Overall Energy Efficiency (percent)	40.0	40.0	38.5	35.7	39.2	35.0	35.3
Coal Consumption (lb/kWh)	0.79	0.79	0.99	1.39	0.81	0.90	1.08
Plant Capital Cost ($ million)	1073	2316	1078	1093	1135	1566	1619
Plant Capital Cost ($/kWe)	1896	2048	1908	1943	1513	1241	1223
Cost of Electricity, Capacity Factor = 0.65							
Capital (mills/kWh)	60.0	64.8	60.3	61.4	47.8	39.2	38.7
Fuel (mills/kWh)	7.3	7.3	7.5	8.1	7.4	8.3	8.2
Maintenance and operating (mills/kWh)	2.2	2.0	2.2	2.2	2.0	2.9	2.9
Total (mills/kWh)	69.4	74.0	70.0	71.8	57.3	50.4	49.8
Sensitivity							
Capacity factor = 0.50 (total mills/kWh)	88.0	94.1	88.8	90.9	72.2	63.0	62.3
Capacity factor = 0.80 (total mills/kWh)	57.7	61.5	58.3	59.9	47.9	42.5	42.0
Capital Δ = 20 percent (Δ mills/kWh)	12.0	13.0	12.1	12.3	9.6	7.8	7.7
Fuel Δ = 20 percent (Δ mills/kWh)	1.5	1.5	1.5	1.6	1.5	1.7	1.6
Estimated Time for Construction (years)	5	6	5	5	5	5	5
Estimated Date of 1st Commercial Service (year)	1995	1995	1995	1995	1995	1995	1995

*Base case
**Performance only

AFB = Atmospheric fluidized bed
DCT = Dry cooling tower
HBtu = High Btu
Ill. = Illinois
LBtu = Low Btu
Mont = Montana
N.D. = North Dakota
PF = Pressurized furnace
$(PFB)_R$ = Pressurized fluidized bed (recu
WCT = Wet cooling tower

Table 2.5-4

FOLDOUT FRAME 2

PARAMETRIC VARIATIONS FOR TASK I STUDY SUPERCRITICAL CO_2 CYCLE

8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24	25	26	27	28**	29**	30	31	32
1348	760	654	564	566	565	566	563	566	565	566	565	566	566	565	565	566	563	563	561	566	566	565	565	563
PF N.D. LBtu	PF III. #5 HBtu	→	AFB III. #6																					→
→	1600	1350	1200	1350																				→
					→	3.14	2.0	2.7																→
							→	0.03	0.08	0.05			→	0.08	0.05									→
									→	20	60	30	→	60	140	30								→
											→	8	24	→	12									→
															→	0.6	0.9	1.0	0.7					→
																			→	40	115	80		→
--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	Post	--	--
																		→	DCT	WCT				→
			→	10	30	20																		→
																						→	0.85	0.8
→	10	→	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--
→	8	→	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--
→	1750	1200	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--
→	0.85	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--
1348	760	654	564	566	565	566	563	566	565	566	565	566	566	565	565	566	563	563	561	566	566	565	565	563
47.7	51.8	47.7	44.9	49.0	46.5	49.1	43.2	48.0	47.4	47.8	47.6	47.7	47.7	47.2	47.2	48.1	44.3	42.7	47.7	48.1	45.6	46.7	46.0	44.1
35.1	42.5	37.2	37.7	41.3	39.1	41.4	36.1	40.4	39.8	40.2	40.0	40.2	40.1	39.7	39.7	40.5	37.1	35.7	39.8	40.5	38.4	39.3	38.6	37.0
35.1	21.4	18.7	37.7	41.3	39.1	41.4	36.1	40.4	39.8	40.2	40.0	40.2	40.1	39.7	39.7	40.5	37.1	35.7	39.8	40.5	38.4	39.3	38.6	37.0
1.41	1.48	1.69	0.84	0.77	0.81	0.76	0.88	0.78	0.79	0.79	0.79	0.79	0.79	0.80	0.80	0.78	0.85	0.89	0.79	0.78	0.82	0.81	0.82	0.86
1700	1103	970	970	1245	1004	1051	1180	1130	1039	1054	1077	1073	1085	1035	1088	1222	882	810	1083	1068	1073	1145	1117	1159
1261	1452	1482	1718	2197	1776	1855	2096	1997	1837	1863	1904	1897	1919	1830	1924	2159	1565	1439	1930	1887	1897	2026	1977	2056
39.9	45.9	46.9	54.3	69.5	56.2	58.7	66.3	63.2	58.1	58.9	60.2	60.0	60.7	57.9	60.8	68.3	49.5	45.5	61.0	59.7	60.0	64.1	62.5	65.0
8.3	20.9	23.9	7.7	7.0	7.4	7.0	8.0	7.2	7.3	7.2	7.2	7.2	7.2	7.3	7.3	7.2	7.8	8.1	7.3	7.2	7.6	7.4	7.5	7.8
2.7	1.5	1.4	1.9	2.2	2.2	2.2	2.3	2.2	2.2	2.3	2.1	2.2	2.2	2.1	2.1	2.2	2.3	2.3	2.2	2.2	2.2	2.2	2.2	2.3
50.8	68.2	72.2	63.9	78.7	65.8	67.9	76.6	72.5	67.6	68.4	69.5	69.4	70.1	67.3	70.3	77.6	59.6	55.9	70.5	69.0	69.7	73.6	72.2	75.1
63.5	82.4	86.7	80.7	100.2	83.3	86.2	97.2	92.1	85.7	86.8	88.2	88.0	88.9	85.3	89.2	98.7	75.1	70.3	89.5	87.6	88.3	93.5	91.7	95.3
42.8	59.4	63.1	53.3	65.2	54.9	56.5	63.8	60.3	56.3	56.9	57.9	57.7	58.3	56.1	58.5	64.4	49.9	47.0	58.7	57.4	58.0	61.2	60.1	62.5
8.0	9.2	9.4	10.9	13.9	11.2	11.7	13.3	12.6	11.6	11.8	12.0	12.0	12.1	11.6	12.7	13.7	9.9	9.1	12.4	11.9	12.0	12.8	12.5	13.0
1.7	4.2	4.8	1.5	1.4	1.5	1.4	1.6	1.4	1.5	1.4	1.4	1.4	1.4	1.5	1.5	1.4	1.6	1.6	1.5	1.4	1.5	1.5	1.5	1.6
5	4	4	5	5	5	5	5	5	5	5	5	5	5	5	5	5	5	5	5	5	5	5	5	5
1995	1998	1990	1995	1995	1995	1995	1995	1995	1995	1995	1995	1995	1995	1995	1995	1995	1995	1995	1995	1995	1995	1995	1995	1995

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Table 2.5-5 (Page 1 of 4)

CAPITAL COST DISTRIBUTIONS FOR SUPERCRITICAL CO_2 CYCLE

	CASE NO.	1	2	3	4	5	6	7	8	9	10
MAJOR COMPONENTS											
PRIME CYCLE											
CO2 TURB-GEN	MM$	130.0	260.0	130.0	130.0	130.0	130.0	130.0	130.0	186.0	130.0
CO2 TURB DRIVE-PUMP-COMP	MM$	20.6	41.1	20.6	20.6	20.6	20.6	20.6	20.6	30.4	20.6
RECUPERATOR	MM$	202.4	405.6	202.4	202.4	202.4	202.4	207.4	202.4	200.0	202.4
PRECOOLER	MM$	10.0	20.0	10.8	11.7	11.5	9.9	10.9	21.2	11.2	8.4
PRIMARY HEAT INPUT AND FUEL SYSTEM											
FURNACE MODULES	MM$	67.0	134.2	66.7	73.8	75.2	54.5	54.1	54.8	71.0	68.2
HIGH TEMP AIR PREHEATER	MM$	7.1	14.2	7.4	7.7	0.	0.	0.	0.	0.	0.
LOW TEMP AIR PREHEATER	MM$	2.1	4.1	2.1	2.4	0.	0.	0.	0.	0.	0.
PRESSURIZING GAS TURBINE (COMP-GEN-HEAT EXCH)	MM$	0.	0.	0.	0.	28.1	64.9	69.0	71.3	18.1	8.2
GASIFIER (INCLUDING BOOST STEAM TURB-COMP	MM$	0.	0.	0.	0.	0.	209.8	228.1	254.3	0.	0.
SUB-TOTAL OF MAJOR COMPONENTS	MM$	439.1	879.2	440.0	448.6	467.8	692.1	715.1	754.6	516.7	437.8
BALANCE OF PLANT											
COOLING TOWER	MM$	1.9	3.8	1.9	1.9	1.9	1.9	1.9	1.9	1.9	1.9
ALL OTHER	MM$	115.9	232.0	117.5	117.5	117.1	127.1	129.8	134.5	106.0	106.0
SITE LABOR	MM$	32.5	64.2	32.7	32.7	36.6	39.3	40.5	42.6	29.6	29.6
SUB-TOTAL OF BALANCE OF PLANT	MM$	150.3	300.0	152.1	152.1	155.6	168.3	172.2	179.0	137.5	137.5
CONTINGENCY	MM$	117.9	235.8	118.4	120.1	124.7	172.1	177.4	186.7	130.8	115.1
ESCALATION COSTS	MM$	169.1	404.7	169.8	172.3	178.8	246.7	254.5	267.8	152.3	134.0
INTEREST DURING CONSTRUCTION	MM$	196.7	498.5	197.6	200.4	208.0	287.1	296.0	311.5	165.5	145.6
TOTAL CAPITAL COST	MM$	1073.1	2318.2	1077.9	1093.5	1134.8	1566.2	1615.2	1699.6	1102.9	969.9
MAJOR COMPONENTS COST	$/KWE	776.3	776.9	778.9	797.4	623.7	548.6	541.7	560.0	680.3	669.3
BALANCE OF PLANT	$/KWE	265.7	265.1	269.2	270.3	207.4	133.4	130.4	132.8	181.0	210.2
CONTINGENCY	$/KWE	208.4	208.4	209.6	213.5	166.2	136.4	134.4	138.6	172.3	175.9
ESCALATION COSTS	$/KWE	298.9	357.6	300.6	306.2	238.4	195.6	192.8	198.7	200.6	204.8
INTEREST DURING CONSTRUCTION	$/KWE	347.7	440.5	349.7	356.3	277.3	227.6	224.3	231.2	217.9	222.5
TOTAL CAPITAL COST	$/KWE	1897.0	2048.4	1908.0	1943.8	1513.1	1241.5	1223.6	1261.2	1452.1	1482.7

Table 2.5-5 (Page 2 of 4)

CAPITAL COST DISTRIBUTIONS FOR SUPERCRITICAL CO_2 CYCLE

	CASE NO.	11	12	13	14	15	16	17	18	19	20
MAJOR COMPONENTS											
PRIME CYCLE											
CO2 TURB-GEN	MM$	85.0	130.0	130.0	127.0	130.0	130.0	130.0	130.0	130.0	130.0
CO2 TURB DRIVE-PUMP-COMP	MM$	13.0	20.1	21.1	22.8	17.9	20.1	21.7	20.6	21.0	20.7
RECUPERATOR	MM$	193.6	299.2	157.3	196.8	262.4	243.2	172.8	196.8	204.0	202.4
PRECOOLER	MM$	9.0	11.2	11.2	10.3	11.0	11.5	10.1	10.1	10.0	10.1
PRIMARY HEAT INPUT AND FUEL SYSTEM											
FURNACE MODULES	MM$	72.3	64.3	72.4	61.6	66.6	56.9	76.9	62.4	67.2	67.0
HIGH TEMP AIR PREHEATER	MM$	7.6	6.9	7.3	6.9	7.9	7.1	7.2	7.1	7.1	7.1
LOW TEMP AIR PREHEATER	MM$	2.2	2.0	2.1	2.0	2.3	2.0	2.1	2.1	2.1	2.1
PRESSURIZING GAS TURBINE (COMP-GEN-HEAT EXCH)	MM$	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.
GASIFIER (INCLUDING BOOST STEAM TURB-COMP	MM$	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.
SUB-TOTAL OF MAJOR COMPONENTS	MM$	382.7	533.8	401.4	427.4	498.0	470.9	420.8	429.0	441.4	439.3
BALANCE OF PLANT											
COOLING TOWER	MM$	1.9	1.9	1.9	1.9	1.9	1.9	1.9	1.9	1.9	1.9
ALL OTHER	MM$	116.0	116.0	116.0	116.0	116.0	116.0	116.0	116.0	116.0	116.0
SITE LABOR	MM$	32.1	32.1	32.1	32.1	32.1	32.1	32.1	32.1	32.1	32.1
SUB-TOTAL OF BALANCE OF PLANT	MM$	150.0	150.0	150.0	150.0	150.0	150.0	150.0	150.0	150.0	150.0
CONTINGENCY	MM$	106.5	136.8	110.3	115.5	129.6	124.2	114.2	115.8	118.3	117.9
ESCALATION COSTS	MM$	152.8	196.1	158.1	165.6	185.9	178.1	163.7	166.1	169.6	169.0
INTEREST DURING CONSTRUCTION	MM$	177.7	228.2	184.0	192.7	216.2	207.2	190.4	193.2	197.3	196.6
TOTAL CAPITAL COST	MM$	969.7	1244.8	1003.7	1051.1	1179.7	1130.3	1039.1	1054.1	1076.6	1072.9
MAJOR COMPONENTS COST	$/KWE	678.1	942.5	710.6	754.6	884.8	832.3	744.3	758.5	780.4	776.8
BALANCE OF PLANT	$/KWE	265.8	264.9	265.6	264.9	266.5	265.1	265.3	265.2	265.3	265.2
CONTINGENCY	$/KWE	188.8	241.5	195.2	203.9	230.3	219.5	201.9	204.7	209.2	208.4
ESCALATION COSTS	$/KWE	270.7	346.3	280.0	292.4	330.2	314.8	289.6	293.6	300.0	298.9
INTEREST DURING CONSTRUCTION	$/KWE	314.9	402.8	325.7	340.2	384.2	366.2	336.9	341.6	349.0	347.7
TOTAL CAPITAL COST	$/KWE	1718.2	2197.9	1777.0	1855.9	2096.0	1997.8	1837.9	1863.6	1904.0	1897.0

Table 2.5-5 (Page 3 of 4)

CAPITAL COST DISTRIBUTIONS FOR SUPERCRITICAL CO_2 CYCLE

	CASE NO.	21	22	23	24	25	26	27	28	29	30
MAJOR COMPONENTS											
PRIME CYCLE											
CO2 TURB-GEN	MM$	130.0	130.0	130.0	130.0	130.0	130.0	130.0	130.0	130.0	141.0
CO2 TURB DRIVE-PUMP-COMP	MM$	20.8	21.8	21.8	25.6	16.6	12.8	20.6	20.6	20.6	20.0
RECUPERATOR	MM$	202.4	174.4	208.8	271.2	100.8	56.8	202.4	202.4	202.4	228.0
PRECOOLER	MM$	9.0	11.4	10.1	11.6	10.1	10.2	10.3	10.4	10.3	11.5
PRIMARY HEAT INPUT AND FUEL SYSTEM											
FURNACE MODULES	MM$	74.8	71.5	67.7	73.5	67.3	74.8	64.0	64.0	67.0	69.0
HIGH TEMP AIR PREHEATER	MM$	7.1	7.2	7.2	7.1	7.7	7.9	7.1	7.1	7.1	7.3
LOW TEMP AIR PREHEATER	MM$	2.1	2.1	2.1	2.0	2.2	2.3	2.1	2.1	2.1	2.1
PRESSURIZING GAS TURBINE (COMP-GEN-HEAT EXCH)	MM$	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.
GASIFIER (INCLUDING BOOST STEAM TURB-COMP	MM$	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.
SUB-TOTAL OF MAJOR COMPONENTS	MM$	446.2	418.4	447.6	521.0	334.7	294.9	436.5	436.6	439.4	478.9
BALANCE OF PLANT											
COOLING TOWER	MM$	1.9	1.9	1.9	1.9	1.9	1.9	5.2	1.9	1.9	1.9
ALL OTHER	MM$	116.0	116.0	116.0	116.0	116.0	116.0	118.9	116.0	116.0	116.0
SITE LABOR	MM$	32.1	32.1	32.1	32.1	32.1	32.1	34.5	32.1	32.1	32.1
SUB-TOTAL OF BALANCE OF PLANT	MM$	150.0	150.0	150.0	150.0	150.0	150.0	158.6	150.0	150.0	150.0
CONTINGENCY	MM$	119.2	113.7	119.5	134.2	96.9	89.0	119.0	117.3	117.9	125.8
ESCALATION COSTS	MM$	171.0	163.0	171.4	192.5	139.0	127.6	170.7	168.2	169.1	180.4
INTEREST DURING CONSTRUCTION	MM$	198.9	189.7	199.4	223.9	161.7	148.4	198.5	195.7	196.7	209.8
TOTAL CAPITAL COST	MM$	1085.4	1034.8	1088.0	1221.6	882.3	809.9	1083.3	1067.8	1073.0	1144.9
MAJOR COMPONENTS COST	$/KWE	789.0	740.2	791.9	921.0	594.0	524.2	778.0	771.8	776.9	847.5
BALANCE OF PLANT	$/KWE	265.2	265.4	265.4	265.1	266.2	266.6	282.7	265.2	265.2	265.5
CONTINGENCY	$/KWE	210.8	201.1	211.5	237.2	172.0	158.2	212.1	207.4	208.4	222.6
ESCALATION COSTS	$/KWE	302.4	288.4	303.3	340.2	246.7	226.8	304.2	297.4	298.9	319.2
INTEREST DURING CONSTRUCTION	$/KWE	351.8	335.5	352.8	395.8	287.0	263.9	353.9	346.0	347.7	371.4
TOTAL CAPITAL COST	$/KWE	1919.2	1830.7	1924.8	2159.2	1566.0	1439.7	1931.0	1887.8	1897.0	2026.1

Table 2.5-5 (Page 4 of 4)

CAPITAL COST DISTRIBUTIONS FOR SUPERCRITICAL CO_2 CYCLE

	CASE NO.	31	32
MAJOR COMPONENTS			
PRIME CYCLE			
CO2 TURB-GEN	MM$	130.0	130.0
CO2 TURB DRIVE-PUMP-COMP	MM$	22.3	23.9
RECUPERATOR	MM$	221.6	240.0
PRECOOLER	MM$	10.1	10.2
PRIMARY HEAT INPUT AND FUEL SYSTEM			
FURNACE MODULES	MM$	70.0	72.5
HIGH TEMP AIR PREHEATER	MM$	7.4	7.7
LOW TEMP AIR PREHEATER	MM$	2.1	2.2
PRESSURIZING GAS TURBINE (COMP-GEN-HEAT EXCH)	MM$	0.	0.
GASIFIER (INCLUDING BOOST STEAM TURB-COMP	MM$	0.	0.
SUB-TOTAL OF MAJOR COMPONENTS	MM$	463.5	486.5
BALANCE OF PLANT			
COOLING TOWER	MM$	1.9	1.9
ALL OTHER	MM$	116.0	116.0
SITE LABOR	MM$	32.1	32.1
SUB-TOTAL OF BALANCE OF PLANT	MM$	150.0	150.0
CONTINGENCY	MM$	122.7	127.3
ESCALATION COSTS	MM$	176.0	182.5
INTEREST DURING CONSTRUCTION	MM$	204.7	212.4
TOTAL CAPITAL COST	MM$	1116.8	1158.7
MAJOR COMPONENTS COST	$/KWE	820.8	863.4
BALANCE OF PLANT	$/KWE	265.7	266.2
CONTINGENCY	$/KWE	217.3	225.9
ESCALATION COSTS	$/KWE	311.6	324.0
INTEREST DURING CONSTRUCTION	$/KWE	362.5	376.9
TOTAL CAPITAL COST	$/KWE	1977.9	2056.4

Table 2.5-6

POWER OUTPUT AND AUXILIARY POWER DEMAND FOR BASE CASE AND PARAMETRIC VARIATIONS: SUPERCRITICAL CO_2 CYCLE

CASE NO.		1	2	3	4	5	6	7	8	9	10
PRIME CYCLE POWER OUTPUT	MW	600.0	1200.0	600.0	600.0	600.0	600.0	600.0	600.0	600.0	600.0
BOTTOMING CYCLE POWER OUTPUT	MW	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.
FURNACE POWER OUTPUT	MW	0.	0.	0.	0.	169.0	673.2	732.1	760.0	168.6	62.7
BALANCE OF PLANT AUX. POWER REQ'D.	MW	5.5	10.5	5.6	5.7	5.6	5.4	5.4	5.6	5.2	5.2
FURNACE AUX. POWER REQ'D.	MW	25.8	51.8	26.4	28.7	9.6	0.	0.	0.	0.	0.
TRANSFORMER LOSSES	MW	3.0	6.0	3.0	3.0	3.8	6.4	6.7	6.8	3.8	3.3
NET STATION OUTPUT	MW	565.7	1131.7	564.9	562.6	750.0	1261.5	1320.0	1347.6	759.5	654.1

CASE NO.		11	12	13	14	15	16	17	18	19	20
PRIME CYCLE POWER OUTPUT	MW	600.0	600.0	600.0	600.0	600.0	600.0	600.0	600.0	600.0	600.0
BOTTOMING CYCLE POWER OUTPUT	MW	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.
FURNACE POWER OUTPUT	MW	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.
BALANCE OF PLANT AUX. POWER REQ'D.	MW	5.6	5.6	5.6	5.6	5.6	5.6	5.6	5.6	5.6	5.6
FURNACE AUX. POWER REQ'D.	MW	27.1	25.1	26.6	25.1	28.6	25.7	26.1	25.8	26.0	25.9
TRANSFORMER LOSSES	MW	3.0	3.0	3.0	3.0	3.0	3.0	3.0	3.0	3.0	3.0
NET STATION OUTPUT	MW	564.3	566.3	564.8	566.3	562.8	565.7	565.3	565.7	565.4	565.6

CASE NO.		21	22	23	24	25	26	27	28	29	30
PRIME CYCLE POWER OUTPUT	MW	600.0	600.0	600.0	600.0	600.0	600.0	600.0	600.0	600.0	600.0
BOTTOMING CYCLE POWER OUTPUT	MW	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.
FURNACE POWER OUTPUT	MW	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.
BALANCE OF PLANT AUX. POWER REQ'D.	MW	5.6	5.6	5.6	5.6	5.6	5.6	10.1	5.6	5.6	5.6
FURNACE AUX. POWER REQ'D.	MW	25.9	26.2	26.2	25.7	28.0	28.9	25.9	25.8	25.8	26.4
TRANSFORMER LOSSES	MW	3.0	3.0	3.0	3.0	3.0	3.0	3.0	3.0	3.0	3.0
NET STATION OUTPUT	MW	565.6	565.2	565.2	565.7	563.4	562.6	561.0	565.7	565.7	565.1

CASE NO.		31	32
PRIME CYCLE POWER OUTPUT	MW	600.0	600.0
BOTTOMING CYCLE POWER OUTPUT	MW	0.	0.
FURNACE POWER OUTPUT	MW	0.	0.
BALANCE OF PLANT AUX. POWER REQ'D.	MW	5.6	5.6
FURNACE AUX. POWER REQ'D.	MW	26.8	28.0
TRANSFORMER LOSSES	MW	3.0	3.0
NET STATION OUTPUT	MW	564.7	563.4

The major disadvantage of this cycle was the extremely high capital costs of major components. A partial list of these capital costs is shown in Table 2.5-7 for the base case. The major cost items are in the heat exchange equipment and power drive turbine.

Table 2.5-7

MAJOR COMPONENTS, CAPITAL COST

Major Components (Partial List)	Capital Costs ($/kW)
Power Turbine	229
Auxiliary Turbine-Pump-Compressor	36
High-Temperature Recuperator	293
Furnace	117
High-Temperature Air Preheater	12

The turbine costs are extremely high because all of the expansion and work output occurs at high pressures and temperatures. This is unlike the steam turbine, where less than 20 percent of the output is derived from the high-pressure turbine stages.

Since high efficiency is the major advantage of this cycle, it is instructive to examine the effects of the important parametric variables of the cycle efficiency. The cycle thermodynamic efficiency and specific power output are shown in Figure 2.5.2 as a function of turbine inlet temperature, in Figure 2.5-3 as a function of total cycle pressure drop, in Figure 2.5-4 as a function of recuperator pinch-point temperature, in Figure 2.5-5 as a function of turbine pressure ratio and in Figure 2.5-6 as a function of pump flow fraction. The desire to go to higher turbine inlet temperatures and higher pressure ratios in order to achieve higher cycle thermodynamic efficiency must be counterbalanced by the added capital cost in major components to achieve these improved conditions.

The increase in efficiency which is achieved with the recompression cycle is shown in Figure 2.5-6 with a pump flow fraction of 1.0 representative of the basic cycle. A pump flow fraction of 0.7 was employed in the base case.

The post-heat cycle, which employs expansion of the CO_2 in the pump drive turbine as it exits high-temperature recuperator and prior to heat addition in the furnace, was considered for its potential to lower the cost of the primary heat input exchanger as a result of lower pressure levels in this component. The study results indicated that this reduction is not achieved and that the power turbine cost increases because this turbine must now operate on the high-temperature CO_2 as it exits the primary heat input exchanger.

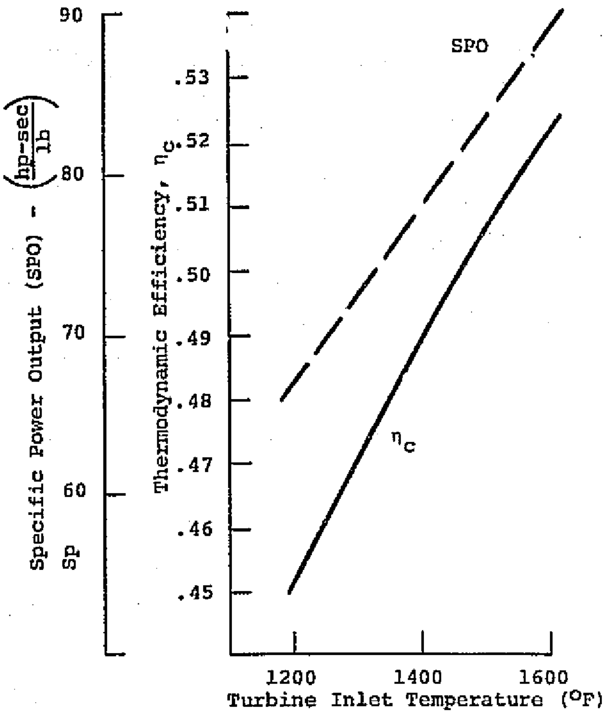


Figure 2.5-2. Effect of Turbine Inlet Temperature on Cycle Performance

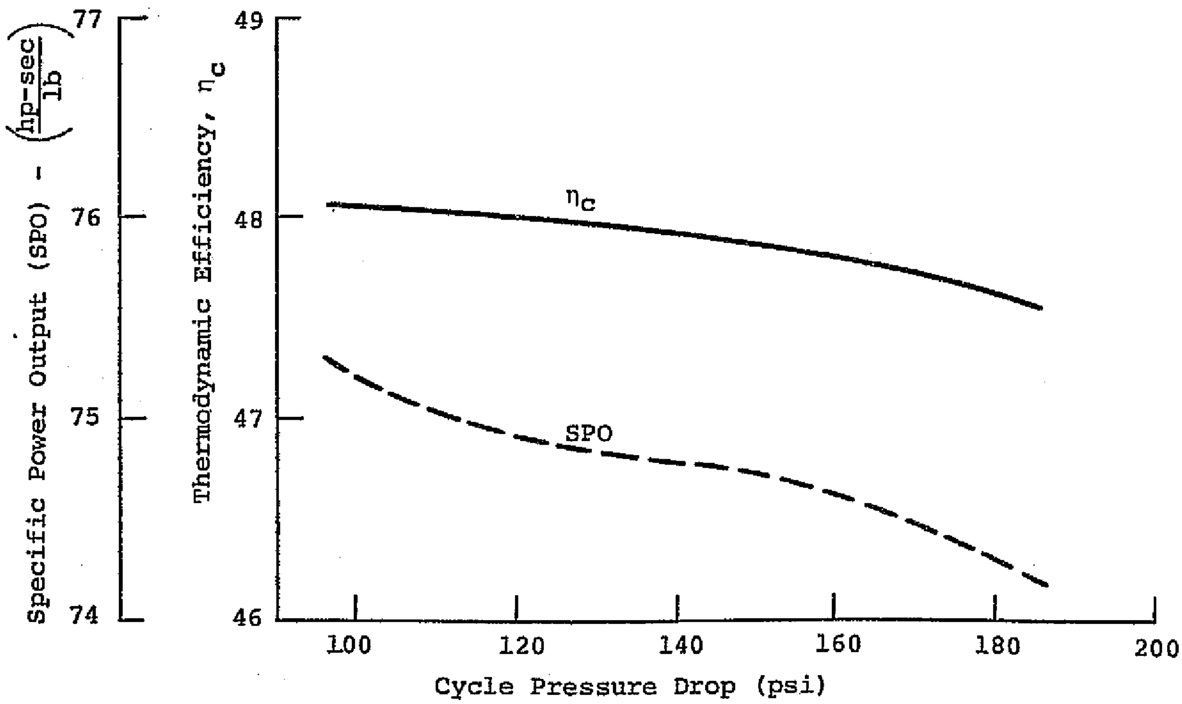


Figure 2.5-3. Effect of Total Loop Pressure Drop on Cycle Performance

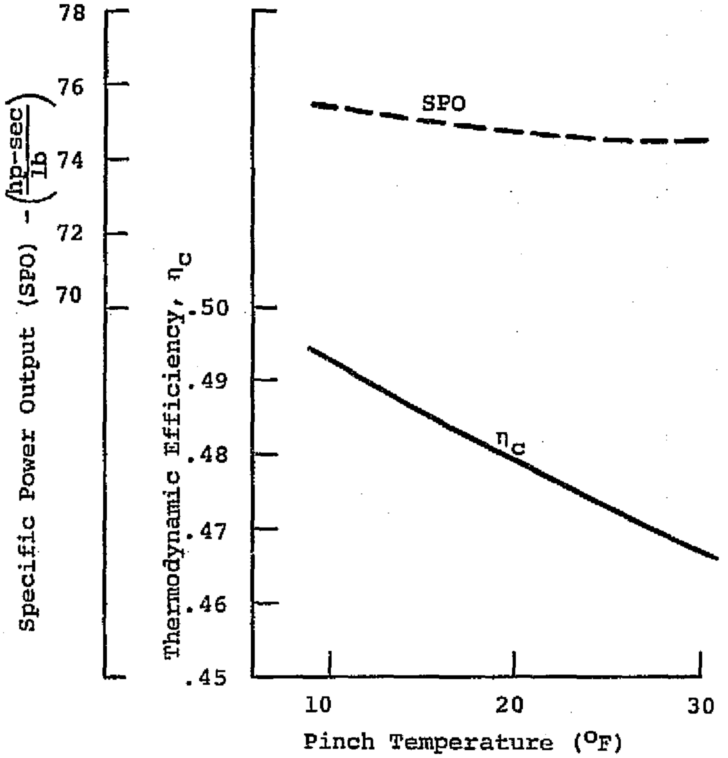


Figure 2.5-4. Effect of Recuperator Pinch-Point Temperature on Cycle Performance

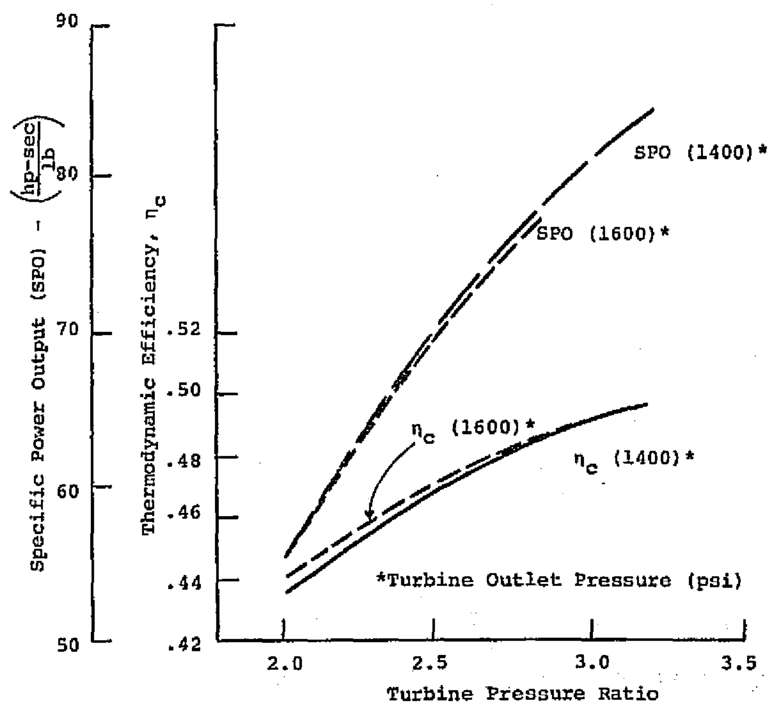


Figure 2.5-5. Effect of Turbine Pressure Ratio on Cycle Performance

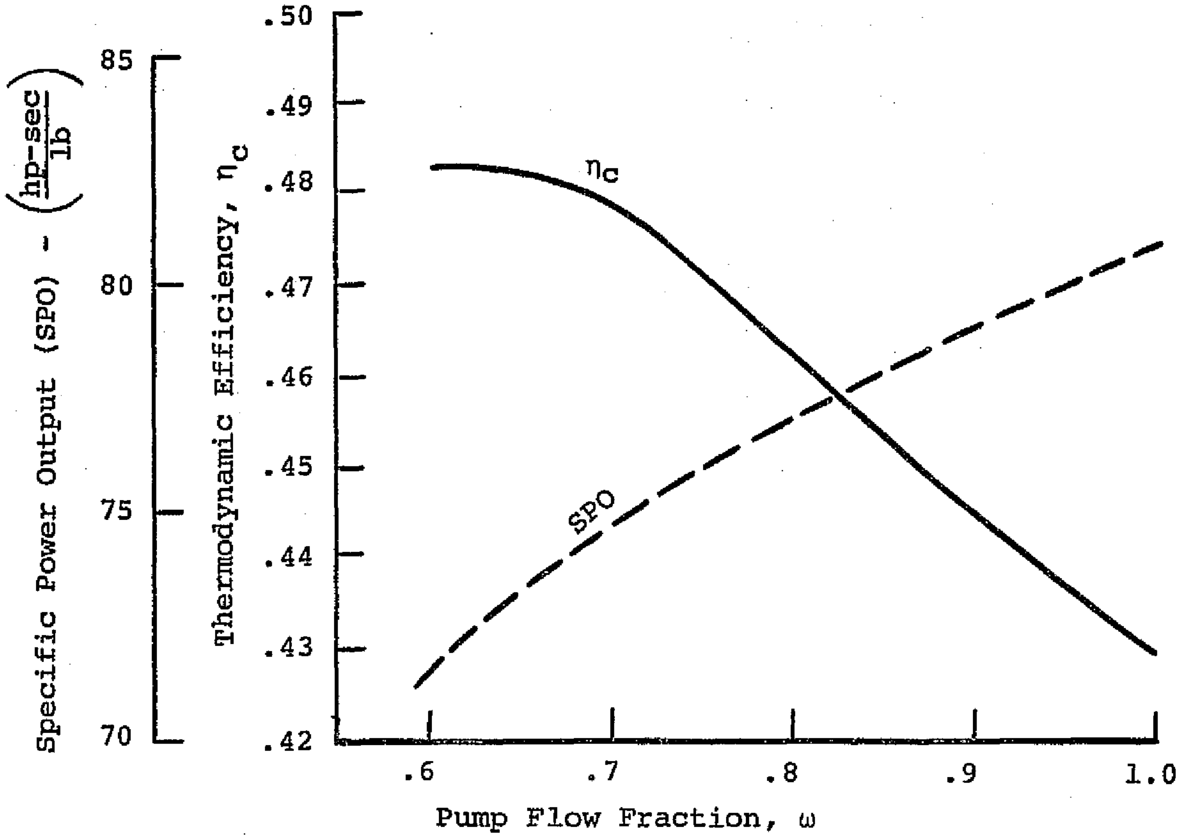


Figure 2.5-6. Effect of Pump Flow Fraction on Cycle Performance

In a final design of the turbine components, cooling of the turbine wheels would probably be necessary. This would require between two and four percent of the compressor flow and result in an overall efficiency decrease of one-quarter to one-half percentage point.

The efficiency of this cycle is not strongly dependent upon flow pressure drop, as noted in Figure 2.5-3. Both the primary heat input and recuperator heat exchangers could benefit from increased allowances in flow pressure drop. A doubling of the shell side pressure drop in the high-temperature recuperator would cause only a 0.25 percentage point decrease in efficiency but would reduce the estimated cost of this unit by more than 10 percent. The same result could take place in the primary heat exchanger: a doubling of pressure drop would decrease the estimated cost of this unit by 4×10^6. Neither of these effects is linear.

Although not studied in this evaluation, it is believed that in order to achieve proper turbine control, 3 percent of the turbine inlet pressure must be made available in the form of control value loss. This effect would cause a reduction in efficiency of 0.3 percentage point.

If the above-mentioned allowances and design improvements were made to the base case, the total capital cost could be reduced by approximately $\$20 \times 10^6$ and the efficiency would drop by approximately 0.5 percentage point. The reduction of power due to efficiency decrease would offset the reduction in capital cost, resulting in approximately the same cost of electricity. Therefore, the efficiencies would be in the upper 30-percent range and capital costs still in excess of \$1800/kW.

RECOMMENDED CASE

The supercritical CO_2 cycle was characterized by efficiencies approximately four to five percentage points greater than conventional steam turbine cycles and capital costs of three times those projected for current steam power plants. With respect to the cost of electricity, the savings in fuel was more than offset by the capital charge, and the resultant cost of electricity was projected to be more than double that for a conventional steam turbine plant.

If the supercritical CO_2 cycle is considered for further study, the base-case configuration which was employed for the Task I Study appears to be an attractive starting point. The recompression cycle with a 2.7/1.0 pressure ratio and turbine inlet temperature of 1350 F (1005 K) would be recommended. A recommendation would be made, however, to allow more flow pressure drop in the heat exchangers in order to reduce capital costs of these components.

REFERENCES

1. Rocketdyne Report R-6805, Office of Naval Research Contract 4507(00), Rocketdyne, Canoga Park, Calif., 1968.

2. Schnurr, N.M., "Heat Transfer to Carbon Dioxide in the Immediate Vicinity of the Critical Point," Transactions of the ASME Journal of Heat Transfer, Vol. 91, February 1969, pp. 16-20.

3. Wood, R.D., and Smith, J.M., "Heat Transfer in the Critical Region—Temperature and Velocity Profiles in Turbulent Flow," American Institute of Chemical Engineers Journal, Vol. 10, March 1964, pp. 180-186.

2.6 ADVANCED STEAM CYCLE

DESCRIPTION OF CYCLE

Many of the recently installed fossil-fired large steam power plants today utilize 800 MW, 3500 psi (2.42×10^7 N/m^2), 1000 F (811 K) steam with reheat to 1000 F (811 K). A few smaller steam plants use 1050 F (839 K) steam, and two units have operated briefly at 1200 F (922 K) and at 1150 F (894 K). The advanced steam plants of this evaluation featured increased initial steam conditions to enhance their efficiency.

The steam power plant uses regenerative feedwater heating to substantially heat the condenser discharge water. This feedwater is heated further in the steam generator by a section called the economizer. The economizer is the last section of the boiler gas path and serves to reduce the boiler gas temperature as low as possible. The gas may be as hot at 750 F (672 K) leaving the economizer. Next the air preheater cools the exhaust gas to stack temperatures of 250 F (394 K) to 300 F (422 K).

The turbine represents a continuous gas path, although it is manufactured in discrete units with their own shells and bearings. The high-pressure section reduces the pressure by five to one and regulates the total steam flow. Its exhaust returns to the reheater except for a small flow which goes to the highest temperature feedwater heater. The reheated steam expands through the intermediate-pressure turbine and then goes to the several final turbine sections. Steam is extracted at many points along the turbine to progressively heat the feedwater. The feedwater pump discharges at a pressure 25 percent in excess of the throttle pressure and is driven by an auxiliary steam turbine.

The major advantages of the steam cycle are the very small pumping power (on the order of 2.5 percent of turbine output) and the near to Carnot processes achieved by employing condensing for heat rejection and by regenerative feedwater heating. Water as a working fluid is also a major advantage; it provides high heat transfer coefficients in both the boiler and the condenser. It expands without moisture through all turbine sections except the last. It enters the condenser slightly wet, which is ideal for condensation.

Steam Cycle Configuration

The advanced steam cycle used for the base case is shown schematically in Figure 2.6-1, with four modules of atmospheric fluidized bed steam generators. These steam generators are described in Section 6. In this configuration each module has a low temperature air preheater that heats combustion air as the stack gas is cooled from 700 F (644 K) to 300 F (422 K). The electrostatic precipitator is at the 700 F (644 K) temperature level preceding the air preheater. The beds themselves operate at 1550 F (1120 K) to produce main throttle steam and to reheat steam. The condensate is

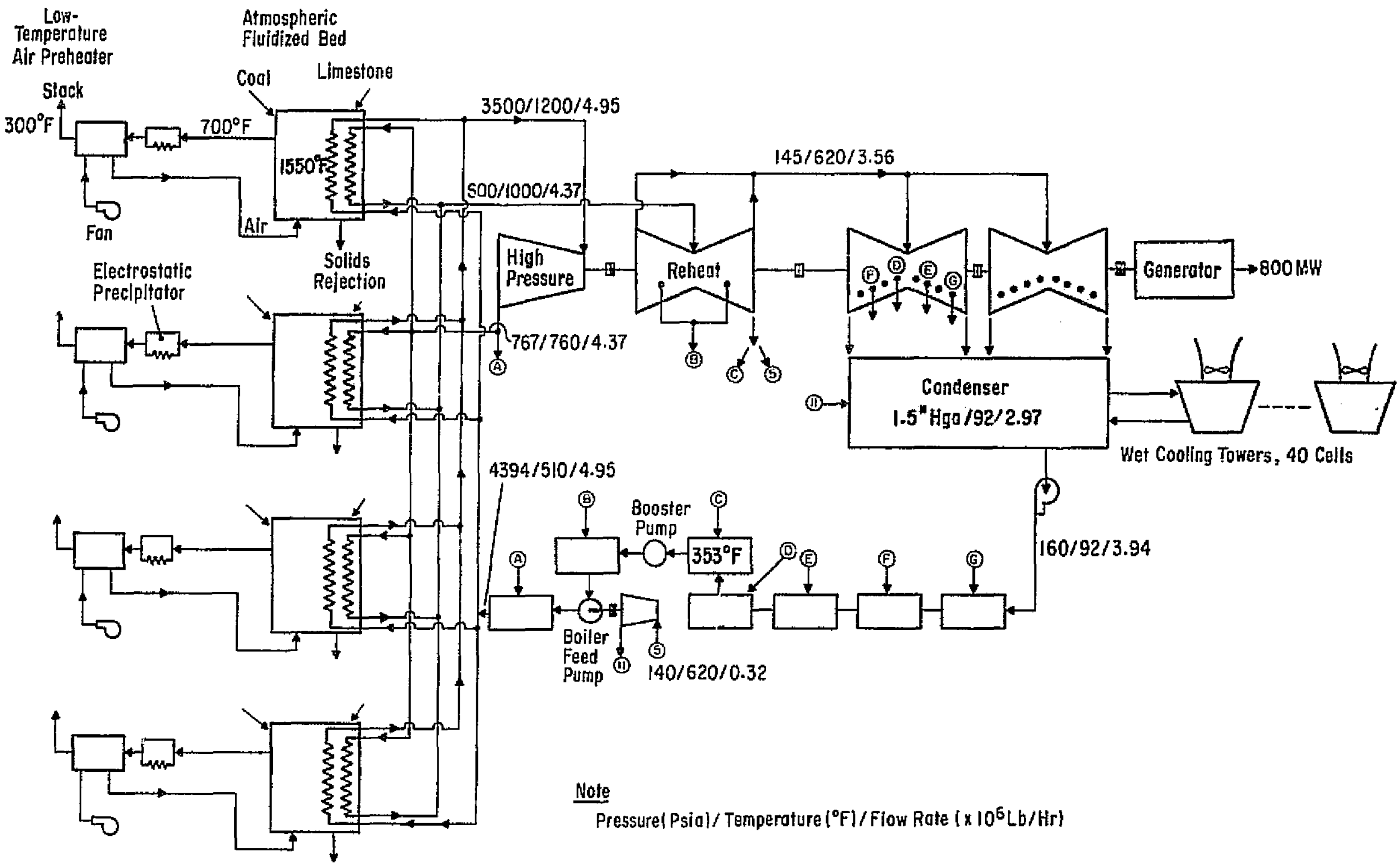


Figure 2.6-1. Advanced Steam Cycle

pumped through a sequence of feedwater heaters and through the boiler feedpump to reach 4394 psia (3.03×10^7 N/m^2), 510 F (539 K) with a water flow of 4.95 million lb/hr (2.25×10^6 kg/hr). This feedwater is subdivided among the steam generators to produce 3500 psig (2.42×10^7 N/m^2), 1200 F (922 K) steam. The high-pressure turbine section reduces the steam pressure to 767 psia (5.29×10^6 N/m^2) at a temperature of 760 F (678 K). A small fraction of this steam flows to the hottest feedwater heater at A; the bulk of the cold reheat steam is returned to the steam generator where it is reheated to 1000 F (811 K) while experiencing a 10 percent pressure drop to 690 psia (4.76×10^6 N/m^2). The reheated steam returns to the second turbine shell. At the exhaust from the reheat turbine shell, steam is extracted for boiler feedpump turbine drive and for feed heating. The bulk of the steam flows through the crossover to the multiple last stage sections. All of the exhaust flows from these last stage sections enter the condenser along with the exhaust from the boiler feedpump turbine. The condensation is effected by the cooling water circulated through wet cooling towers.

The steam turbine cycle is described by the generator output, 800 MW; the configuration, Tandem Compound 4 Flow and 33.5-in. (0.851 m) last stage buckets (LSB) (TC 4F 33.5); the condenser back pressure, 1.5 in. Hga (5.07×10^3 N/m^2); the feedwater temperature, 510 F (539 K); and the guarantee heat rate, 7482 Btu (7.89×10^6 J) of heat added to the steam cycle per kilowatthour produced by the steam cycle (45.6 percent steam cycle efficiency).

The atmospheric fluidized bed is one of four furnace-steam generator systems that were evaluated. In the atmospheric fluidized bed (AFB) cases the combustion gases are cooled to 700 F (644 K) with the 510 F (539 K) feedwater. This avoids the need for a high-temperature air preheater. A low-temperature air preheater reduces stack gases to 300 F (422 K).

The pressurized fluidized bed cases were comparable except that a gas turbine was used to pressurize each furnace with the gas turbine exhaust used to heat feedwater above the level of 232 F (384 K) in place of regenerative steam heating. Gas turbine power was added to the net steam power generation. In one case a recuperator was used on the gas turbine with no feedwater heating from the exhaust gases; the steam cycle was unchanged from the basic AFB case.

The pressure-fired furnace cases burning clean gaseous fuels employed all elements of a combined power system. To make low-Btu fuel gas, a coal gasifier of the fixed-bed type was furnished compressed air from the gas turbines, and steam from a gas turbine heat recovery steam generator. Each gas turbine had a combustor-boiler that provided heat to the basic steam plant and discharged hot gases through the gas turbine to the heat recovery steam generators. For these cases, the aggregate power of the gas turbines and their heat recovery steam turbine exceeded that of the advanced steam plant. Better thermal performance would result from an integration of all the steam turbine components.

The major steam cycle variations about the base case included variation of initial pressure, of initial temperature, and of reheat temperature, addition of a second reheat, and change of condenser conditions and of feedwater temperature.

The heat sources and system configurations were major parmetric case variables (Table 2.6-1).

Table 2.6-1

ADVANCED STEAM CASE VARIABLES

System Parameters	Base Case	Variations
Steam Cycle		
Generator (MW)	800	600,1200,1600
Turbine inlet temperature (°F)	1200	1000,1400
Turbine inlet pressure (psig)	3500	4000,(2400 psig/ 1000 F/1000 F)
Reheat temperature (°F)	1000	1200,1400
Feedwater temperature (°F)	510	560,547,480
Condensing pressure (in. Hg abs)	1.5	1.9,3.45
Heat Source for Steam		
Coal burning	AFB	PFB, PFB_R, Conventional
Low-Btu gas	—	PF
Solvent refined coal	—	Conventional
Coal Used	Ill.#6	N.D. Lignite, Mont. Sub-Bi
Support Gas Turbine Temperature (°F)	—	1800 for LBtu, 1600 for PFB

Note: AFB = Atmospheric fluidized bed
LBtu = Low Btu
Ill. = Illinois
Mont. = Montana
N.D. = North Dakota
PF = Pressurized furnace
PFB = Pressurized fluidized bed
PFB_R = Pressurized fluidized bed, recuperative
Sub-Bi = Sub-Bituminous

RATIONALE FOR POINT VARIATIONS

The base case for advanced steam uses ratings and conditions that are typical of the largest fossil-fired power plants, except for the advance from 1000 to 1200 F (811 to 922 K) for the turbine throttle temperature. Such a design for 800 MW would be a distinct challenge both for the turbine and for the boiler. The atmospheric fluidized bed with its peak temperature at 1550 F

(1120 K) offers potential advantages in design concept over the conventional furnace. The reheat at 1000 F (811 K) is conventional as is use of feedwater heating at the temperature corresponding to exhaust from the high-pressure turbine. A second reheat would result in superheated steam flowing into the condenser. This is not conventional for condensing steam turbines. For this reason single reheat has been used in all cases with 1200 F (922 K) throttle.

The variations in power level examined the economy of rating as well as the limits to steam turbine size in conjunction with 1200 F (922 K) steam. Where single units were not practical, twin units were evaluated at 600 to 800 MW per unit. The atmospheric fluidized bed boiler would comprise multiples of a common module in every instance, so more units rather than a scale-up in size of the furnace module would be used. Principal economies outside the steam turbine could arise from balance-of-plant economies of size.

The alternative 1000 F (811 K) throttle condition is conventional if the reheat is held to 1000 F (811 K). However, consideration was given to reheat in this case to 1200 F (922 K) and to 1400 F (1030 K). The reheater would be at 700 psi (4.83 x 10^6 N/m^2) instead of 3500 psi (2.42 x 10^7 N/m^2), thus simplifying the design of the high-temperature heat input section. A second reheat to 1200 F (922 K) was also evaluated since the steam flow to the condenser would still be wet and not superheated.

Although increased pressure generally accompanies increased temperature in steam plants, the advantage is very slight once a design is above the critical pressure of 3200 psi (2.22 x 10^7 N/m^2). A limited extension to 4000 psi (2.77 x 10^7 N/m^2) was made to determine cost sensitivity to the pressure parameter. As a reference case, a plant at 2400 psi (1.66 x 10^7 N/m^2), 1000 F (811 K) superheat, 1000 F (817 K) reheat was evaluated since less than supercritical steam plants are commonly used for both baseload and mid-range operation at this pressure level.

Most commonly, the high-pressure turbine shell of the steam turbine-generator has no extraction points. The high-pressure feedwater heater draws steam from the high-pressure turbine exhaust resulting in 510 F (539 K) feedwater. Extraction from the high-pressure turbine was evaluated resulting in 560 F (566 K) feedwater. The condenser back pressure variations were chosen so that performance in the extremes of ambient conditions could be evaluated as well as determination of performance with dry cooling towers.

The alternative fuels fired directly affect the design of the atmospheric fluidized bed. Conventional furnace designs were evaluated for firing of the three coals as well as the liquefied fuel.

The three low-Btu gas fuels were considered only for the pressure-fired boiler. In this instance, there was integration

of the air compressor and of steam produced in the exhaust of the pressure-firing gas turbine with both the coal gasification plant and its own bottoming steam turbine, but not with the advanced steam turbine-generator of the prime cycle.

The pressurized fluidized bed fired with three different coals utilized the gas turbine exhaust to substitute for part of the feed heating train of the steam turbine. In addition a single case explored the alternative use of gas turbine exhaust to heat compressed air for combustion.

ANALYTICAL PROCEDURE AND ASSUMPTIONS

Steam Turbine-Generator

The analytical procedures and the assumptions used in this evaluation are identical to those applied to current steam turbine-generator products. These are outlined in Reference 1. Estimated efficiencies have been found to match those calculated using this Reference, within 0.25 percent. The specific sizes and conditions for most of the advanced turbines evaluated in this study are beyond the capability of current steam turbines in the utilization of high temperatures. However, except for those stated conditions, current constraints such as last stage loadings have been followed. The turbines considered and the assumptions as to their manufacture and performance follow conventional practice, but extend the practice into unproven high-temperature regions where units are not offered for manufacture.

Cycle Configuration

The elemental steam power plant consists of a feedwater pump, a heat source now called a steam generator, a steam expander (formerly a reciprocating steam engine but now a turbine), a steam condenser, and a driven load such as a generator. This elemental plant without embellishments is called a Rankine steam cycle. This simple form of steam cycle has been progressively modified in utility applications in order to produce the most economic electric power generation. Regenerative feedwater heaters have been added and steam reheating. Initial pressures and temperatures have been increased. The large auxiliary power for the boiler feedpump drive, of the order of 35 MW, is provided by low-pressure ratio auxiliary steam turbine drives. Figure 2.6-2 shows the heat balance for a current 1000 MW utility steam cycle. The highly integrated nature of the steam cycle is apparent using a six-flow exhaust. The approximate upper limit for a four-flow unit would be 880 MW. As is customary for such units, the net unit output is divided into the rate of heat input to the steam cycle to express the unit heat rate of 7946 Btu/kWh. The steam cycle efficiency would be 3412.14 divided by the heat rate or 43.0 percent (the inverse of the ratio).

Figure 2.6-3 shows the physical configuration of the steam turbine sections and their combination into a tandem compound

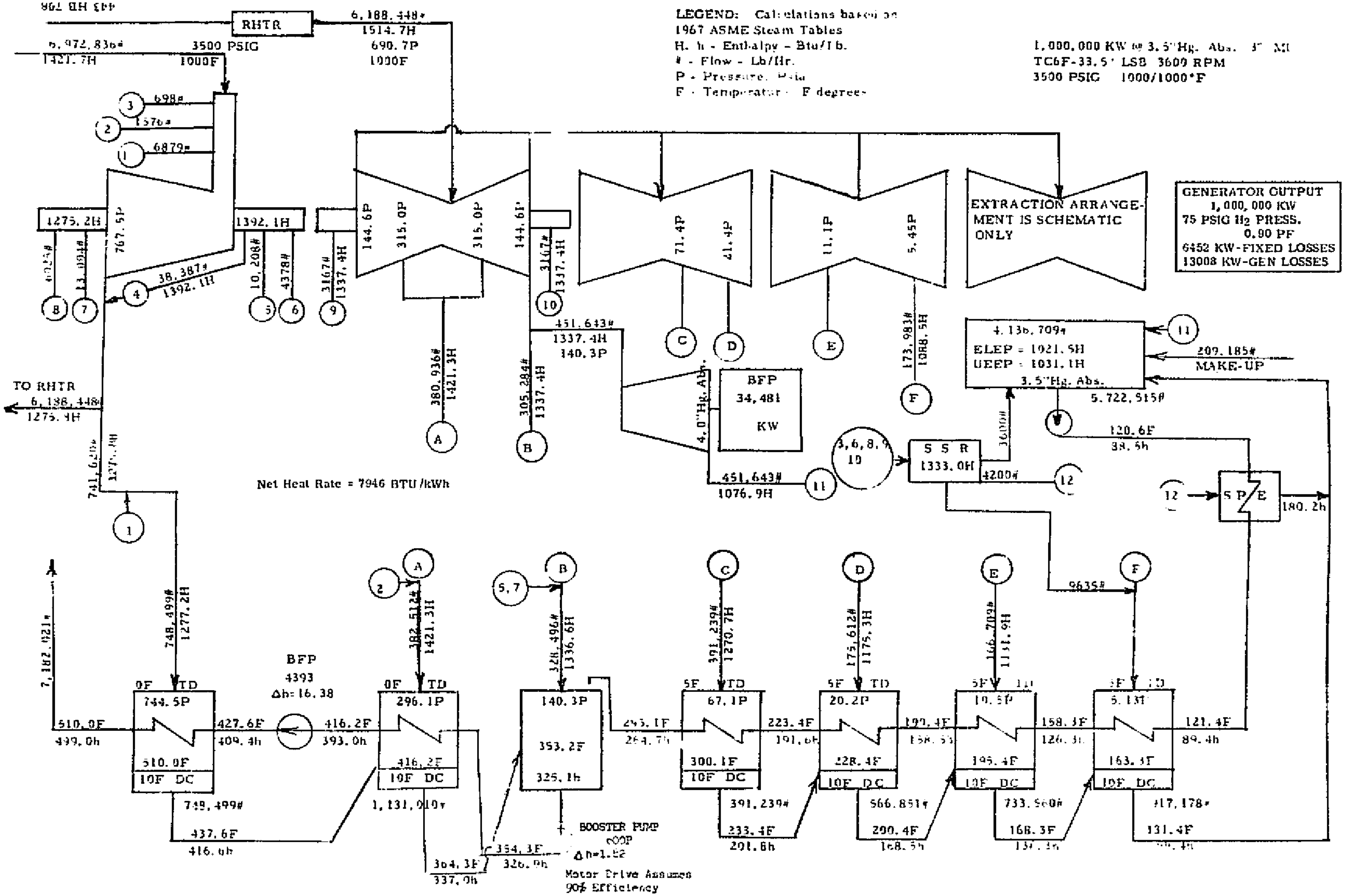


Figure 2.6-2. Conventional Utility Steam Cycle

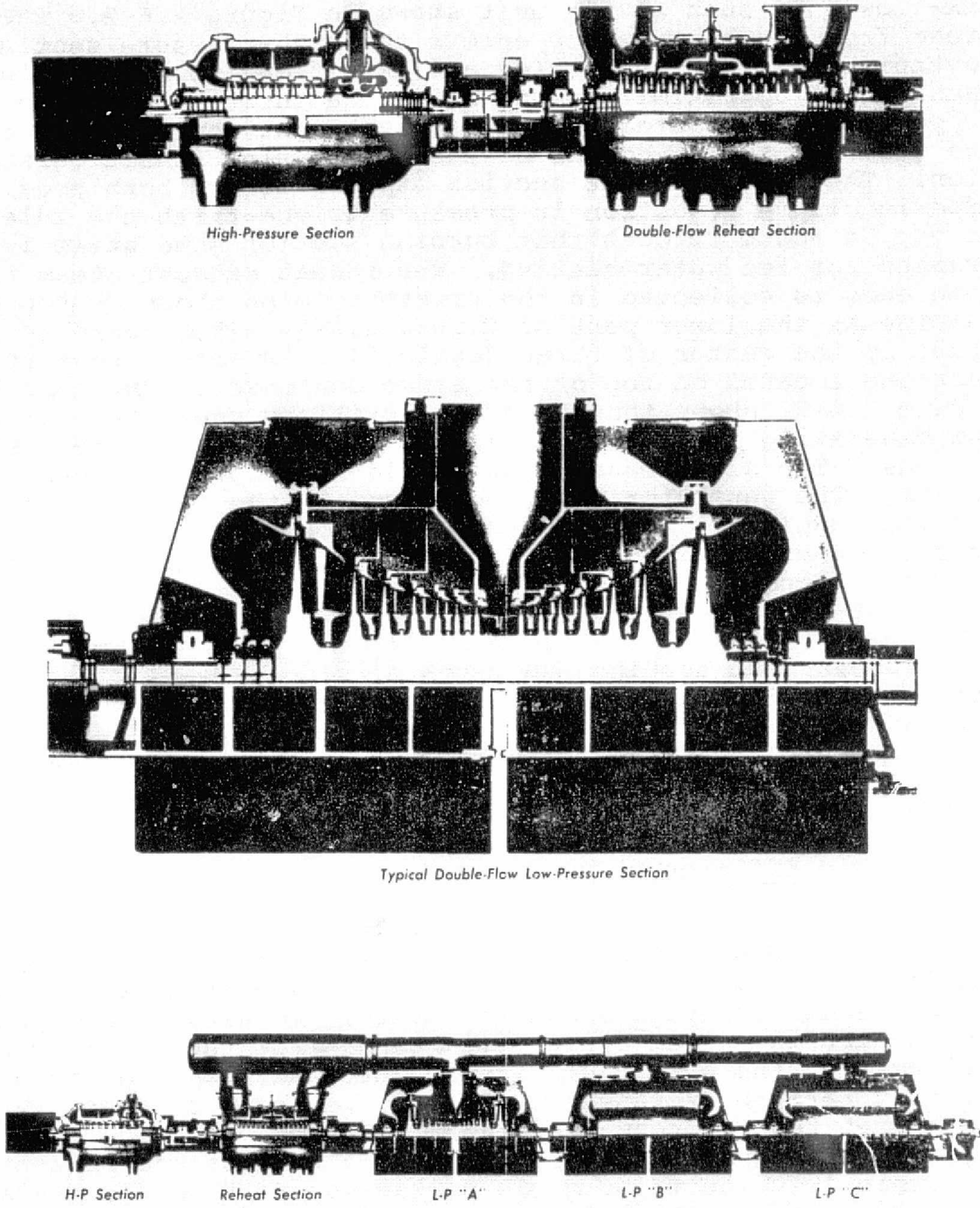


Figure 2.6-3. 3600-Rpm, Tandem-Compound, Six-Flow, Reheat Steam Turbine Rating Range: 550,000 to 1,000,000 kW

six-flow unit such as the unit shown in Figure 2.6-2. The throttle steam from the heat source enters the high-pressure section through governing valves or throttles and then flows toward the left, exhausting at approximately one-fifth the throttle pressure. About 90 percent of the steam returns to the heat source to be reheated and returned to the center of the double-flow reheat turbine section. The steam in that section expands toward both ends, again experiencing a reduction in pressure to one-fifth the inlet pressure. En route through that turbine section some steam is extracted for feedwater heating. The reheat exhaust steam from both ends is collected in the crossover pipe shown on top of the turbine in the lower part of Figure 2.6-3. The crossover feeds steam to the center of three double-flow low-pressure turbine sections located on top of the steam condenser. The heat balance, Figure 2.6-2, shows that both the auxiliary steam turbine and the deaerating feedwater heater use steam at the crossover conditions. The six exhaust flows would flow downward into the condenser. The generator would be bolted to the extreme right steam turbine coupling. In this study six-flow low-pressure turbines (three low-pressure double-flow sections) were used for the 1200 MW Case 3, and for the double reheat Case 14. All other turbines were four-flow low-pressure turbines (two double-flow sections).

The largest turbine buckets are used at the last stage; last stage bucket length is a significant steam turbine characteristic. All but one of the cases evaluated used 33.5-in. (0.851 m) last stage buckets. The exception was the high back pressure turbine for use with a dry cooling tower, Case 9, which used 20-in. (0.508 m) last stage buckets.

Steam Cycle Efficiency

The efficiency of the steam cycle is directly influenced by the steam turbine efficiency, by the kinetic energy in the steam leaving the last turbine buckets, and by the arrangement and number of feedwater heaters and reheaters. To clearly and properly distinguish the steam cycle performance from the steam turbine performance, the entire utility industry has adopted the use of net heat rate to express the steam cycle thermal input divided by the net turbine room electrical kilowatt output, or the Btu per kilowatthour. Data for conventional plants are presented in Reference 2. The major cycle variables are throttle pressure and temperature, reheat temperature, condenser back pressure, number of feedwater heaters, and final feedwater temperature. Performance is given for normal guarantee point operation, and for the conventional design condition at valves wide open with 5 percent additional flow and approximately 4 percent additional power generation. Identical design margins are specified for the steam generator output and for the condenser capability. As a result all steam plants are designed for continuous operation at a 5 percent flow margin and approximately 4 percent excess power generation capability. This design practice was followed in the study. The exhaust flow limit for the 33.5-in. (0.851 m) last stage rows for these 3600 RPM turbines would be 992,000 pounds per hour (4.50×10^5 kg/hr).

The steam turbine-generators have been sized to produce their electrical output at 1.5 in. Hga (5.07×10^3 N/m^2), zero percent makeup, while the high back pressure (HBP) unit was sized at 8.0 in. Hga (2.70×10^4 N/m^2), zero percent makeup. All generators are rated at their maximum hydrogen pressure. The type stator cooling is conductor (liquid) cooling. The generators are assumed to operate at rated hydrogen pressure and 0.90 power factor at all load points.

Table 2.6-2 lists the cycle assumptions that were made.

Table 2.6-2

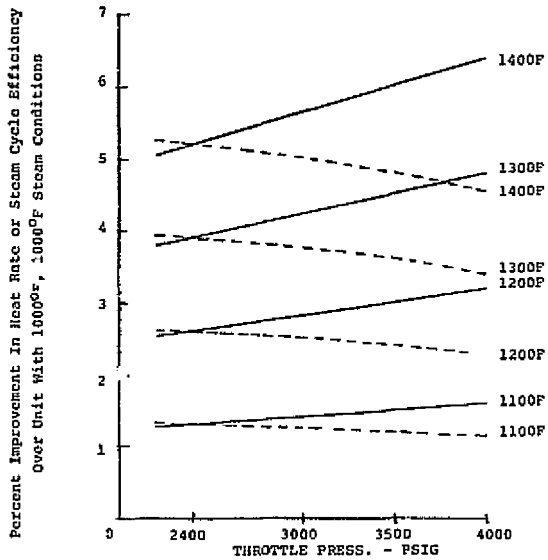
STEAM TURBINE CYCLE ASSUMPTIONS

Components	Assumptions
Boiler feedpump	a. Discharge pressure 1.25 x throttle pressure b. Efficiency 80%
Auxiliary turbine	a. Extracts steam from the main turbine at approximately 150 psia b. Exhausts at a pressure 0.5 in. Hg above main condenser pressure c. Efficiency 78%
High-pressure feedwater heater	a. Normally receives steam from the high-pressure turbine exhaust b. For the HARP cycle Case 5 (Heater Above Reheat Point), the steam is extracted from the high-pressure turbine.
Extraction lines	a. 3% pressure drop

Steam Cycle Effects Due to Pressure and Temperature

Variation of throttle pressure and temperature and of reheat temperature have a dominant effect on steam cycle heat rate and efficiency. For large steam turbine-generators the most common throttle pressures have been 2400 psig (1.66×10^7 N/m^2) and 3500 psig (2.42×10^7 N/m^2), with throttle and reheat temperatures of 1000 F (811 K). Figure 2.6-4 shows the percent improvement over base cycle efficiency or heat rate as the throttle and reheat temperatures are increased up to 1400 F (1030 K). It is notable that increased reheat temperature is 80 percent as effective at 3500 psig (2.42×10^7 N/m^2) as the same increase in throttle temperature. Due to the reduced pressure of reheat 770 to 690 psia (5.31×10^6 to 4.76×10^6 N/m^2), and the absence of throttle controls for reheat, this alternative is economically and technically

preferable to advances in throttle temperature. The general influence due to throttle pressure is a 1 percent improvement going to 4000 psig and a 2 percent poorer heat rate going to 2400 psig (2.77×10^7 N/m^2) from 3500 psig (2.42×10^7 N/m^2).



Notes:
1. Numbers near curves are temperatures in °F
2. Solid lines (———) are throttle temperature
3. Dashed lines (-----) are reheat temperature

Figure 2.6-4. Temperature Effects on Heat Rate for Single Reheat Steam Turbine Cycles

The feedwater temperature and the reheat steam conditions are dependent on throttle conditions. When throttle temperature is increased above the 1000 F (811 K) standard, the enthalpy of the steam returning to the reheater increases 0.5 Btu/lb (1.16×10^3 J/kg) for each degree F increase in throttle temperature. Table 2.6-3 shows these dependencies on throttle pressure at the valves' wide open condition and in parenthesis for the valves' wide open and 5 percent overpressure condition.

Additional Reheating Benefits

The dramatic improvements realized when steam is initially reheated are not extended to subsequent reheating arrangements. Cycle efficiency and heat rate are improved 1.75 percent for addition of a second reheat. No further increase in efficiency results if a third reheat is added. As a result double reheat is the limit of practical exploitation of this avenue to higher efficiency.

Figure 2.6-5 shows the heat rate improvements resulting from increased throttle, first reheat, and second reheat temperatures. These improvements are additive. Again a most practical approach to improved efficiency would be through use of standard throttle conditions, but with the reheat temperatures increased.

Table 2.6-3

COLD REHEAT STEAM CONDITIONS FOR
SINGLE REHEAT UNITS

Steam Conditions (psig/°F/°F)	Final Feedwater Temperature (°F)	Average Cold Reheat Enthalpy (Btu/lb)	High-Pressure Turbine Exhaust Pressure (psia)
2400(2520)/1000/1000	480(485)	1314(1311)	584(613)
3500(3675)/1000/1000	510(516)	1278(1273)	767(805)
4000(4200)/1000/1000	513(518)	1259(1254)	788(830)

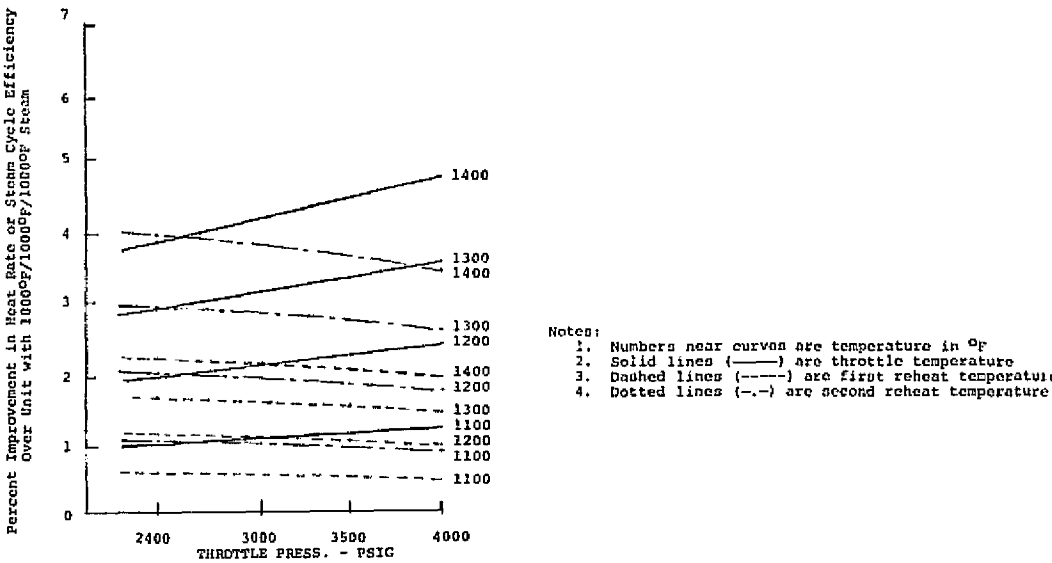


Figure 2.6-5. Pressure-Temperature Effects for Double Reheat Steam Turbine Cycles

DESIGN AND COST BASIS

Materials of Construction, Size and Weight

Materials with sufficient strength for use at 1200 F (922 K) or even to 1400 F (1030 K) for steam turbine designs may be found in materials handbooks such as the ASM Metals Handbook or the Aerospace Structural Metals Handbook. None of these materials have ever been cast or forged into the shapes and sizes required for large steam turbines. Table 2.6-4 indicates the approximate weights and sizes of the major turbine components for a 1000 MW unit. Approximate dimensions are provided to help visualize the physical size of the components in a steam turbine-generator.

Table 2.6-4

COMPONENT WEIGHTS AND SIZES

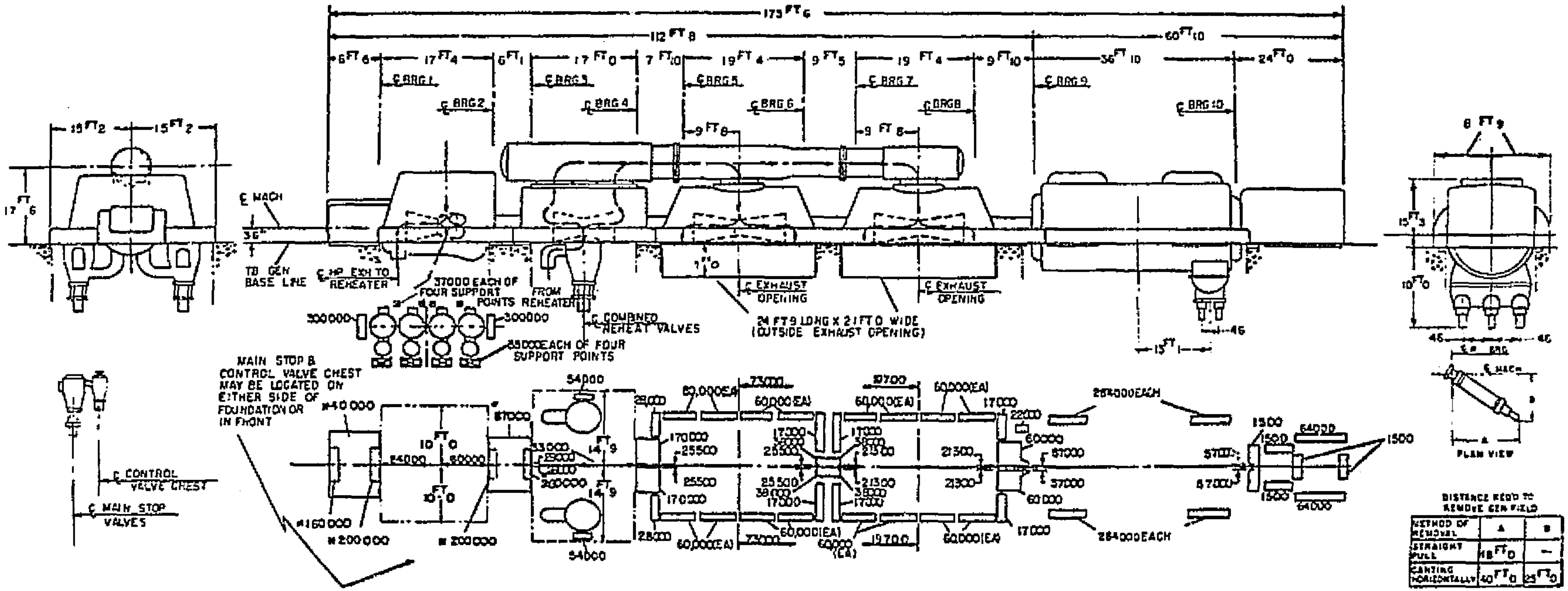
Component	Approximate Weight (lb)	Approximate Size
Main valves & piping	550,000	—
HP-shells (inner & outer) & diaphragms	350,000	22'(L)x15'(W)x10'(H)
HP-rotor	50,000	25'x40" diameter
Intercept valves (two per unit)	120,000	—
Reheat-shells (inner & outer) & diaphragms	300,000	22'(L)x15'(W)x8'(H)
Reheat-rotor	50,000	25'x40" diameter
LP casings (inner & outer) & diaphragms	900,000 (per LP section)	24'(L)x18'(W)x10'(H)
LP-rotor	290,000	27'x30" diameter
Generator (total)	1,400,000	—

It can be seen from Table 2.6-4 that the pieces to be manufactured are very large; they are also very complex in shape. Figure 2.6-6 is an outline drawing for a typical large, single reheat unit showing the approximate sizes of the major components and foundation loadings that can be expected.

In the evaluation of materials for 1200 F (922 K) operation, materials presently available with the required strength were selected, and then the problems that may be encountered in using these materials were identified. The improvement in materials required to advance steam temperatures substantially beyond 1000 F (811 K) would require cooperative development efforts with one or more large steel mills. It would require substantial steel mill investments in increased forging press and furnace capacity. Based upon past experience, a program to develop a satisfactory high-temperature rotor would take at least ten years from the initiation of the project to the first application. Additional in-service operating experience would be required before the material development program could be considered complete.

Steam Turbine-Generator Costs

The prices for conventional steam turbine-generators were determined using the General Electric Apparatus Handbook. The pricing methods are explicitly detailed in the Handbook including



NOTES

1 STEAM CONDITIONS 3500 PSIG-1000/1000 °F REHEAT WITH 33½ IN LAST STAGE BUCKETS

2 HEAVIEST PIECE DURING ERECTION - (GENERATOR STATOR) 760000 LBS
HEAVIEST PIECE AFTER ERECTION - (GENERATOR FIELD ROTOR 160000 LBS

*3 160000 & 200 000 LB LOADINGS DO NOT OCCUR SIMULTANEOUSLY BUT ARE SUPERIMPOSED ON 40000 LB LOAD 160000 LB LOAD IS ZERO FOR NORMAL OPERATION FOR FULL VALUE OF 160000 LB LOAD 200000 LB LOAD IS ZERO

4 THE COMBINED REHEAT VALVE ARRANGEMENT INCLUDES THE REHEAT STOP AND INTERCEPT VALVES.

5 SHAFT ROTATION IS CLOCKWISE WHEN VIEWED FROM COLLECTOR END OF GENERATOR

6. LOAD VALUES SHOWN ARE FOR TURBINE-GENERATOR UNIT ONLY ADDITIONAL LOAD CONTRIBUTED BY CONDENSER NOT INCLUDED

7. DISTANCE REQUIRED TO REMOVE MAJOR PARTS - NOT INCLUDING SLINGS

A GENERATOR STATOR - 14 FT 3 FROM TB-GEN. BASE LINE WITHOUT TERMINAL BOX ATTACHED

B GENERATOR COOLERS - 19 FT 0 IN HORIZONTALLY FROM VERTICAL ℄ OF GEN.

8. THIS OUTLINE SHALL NOT BE USED FOR CONSTRUCTION PURPOSES THE GENERAL EQUIPMENT ARRANGEMENT, DIMENSIONS AND LOADINGS ARE TO BE CONSIDERED PRELIMINARY AND SUBJECT TO LATER MODIFICATION

9. RECOMMENDED MINIMUM CRANE HOOK HEIGHT ABOVE TB. GEN. BASE LINE - 33 FT 7 IN

10 RECOMMENDED MINIMUM FOUNDATION HEIGHT - 40 FT 0 IN

11. THE HORIZONTAL FORCES ACTING AT THE KEYS INCLUDE A SEISMIC FACTOR OF 0.120.

PRELIMINARY OUTLINE
STEAM TURBINE - GENERATOR UNIT

TC4F - 33.5" lsb

Figure 2.6-6. Outline Drawing for a Typical Large, Single Reheat Unit

differentials for standard and for optional features of large steam turbine generators. Figure 2.6-7 illustrates the principal factors determining cost. The MW rating is not the designated output of the unit as used in this study. It is a smaller number that prorates the output to conditions of 3.5 in. Hga (1.18×10^4 N/m^2) condenser back pressure and 3 percent makeup waterflow through the feedwater system. The price base point is indicated for each unit based on the configuration of the condensing turbine sections. Added or reduced output is realized at a constant factor leading to the uniform characteristic slopes. The limit on last stage steam flow produces the approximate cutoff point at the extreme right of each curve. To this base price is added the indicated values at the top of the figure that relate to pressure at the throttle and the extraction of steam for boiler feedpump drive.

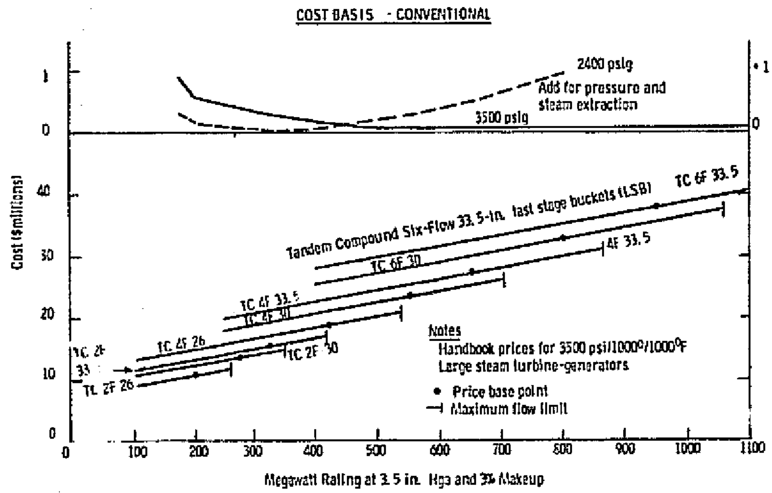


Figure 2.6-7. Cost Basis—Conventional Steam Turbine-Generators

The method employed for determining the cost of unconventional units depends upon the establishment of an accurate description of the entire unit. This was accomplished by actually beginning the design of the steam path and shells for the turbine. The new design was then compared with a conventional design and all significant differences were identified. These differences were then evaluated as to how they would affect the cost of the turbine-generator.

Some of the major differences analyzed for this study were the use of new, untried materials for the major turbine components, and the additional labor required because of longer machining times. Table 2.6-5 lists the estimated costs for a tandem compound six-flow 33.5-in. last stage bucket (0.851 m) base unit with standard accessories. The base rating is 950,000 kW and 1,140,000 kVA rated at 3.5 in. Hga (1.18×10^4 N/m^2) and 3 percent makeup. Steam throttle pressure is 3500 psig (2.42×10^7 N/m^2) while temperatures were as indicated. These estimated prices reflect the best current judgment as to what the price of

a turbine with 1200 F (922 K) temperatures might cost. However, this study did not include full development of either a design or materials evaluation. Further development and additional analysis would be required to obtain more definitive prices.

Table 2.6-5

ESTIMATED STEAM TURBINE PRICES

(950 MW, TC 6F 33.5)*

Alternatives	Steam Temperatures (°F/°F)	Estimated Price ($ millions)
1	1000/1000	31
2	1200/1000	77
3	1000/1200	57
4	1200/1200	103

*950 Megawatt, Tandem Compound, 6 Flow, 33.5 in. LSB

A similar basis for pricing 1400 F (1030 K) steam turbines could not be made. The prices used for 1400 F (1030 K) were price extrapolations and have no technical basis for costing.

These turbine-generator costs are illustrated in Figure 2.6-8 including the resulting cost for a Tandem Compound 4 Flow 33.5-in. (0.851 m) last stage bucket (TC 4F 33.5) steam turbine-generator.

RESULTS

The base case was a 3500 psig, 1200 F (2.42×10^7 N/m^2, 922 K) steam turbine-generator with 1000 F (811 K) reheat serviced by four atmospheric fluidized bed steam generator modules. The total performance of this base case is presented in the summary of Table 2.6-6. The generator output of 800 MW was reduced to a net plant output of 745 MW by the various auxiliary demands as follows:

Furnace module power	31.9 MW,	4	percent of 800 MW
Balance of plant power	19.0 MW,	2.4	percent of 800 MW
Transformer loss power	3.8 MW,	0.5	percent of 800 MW
Net station output power	745.3 MW,	93.2	percent of 800 MW

Presentation of Results

The parametric variations evaluated and their economic and thermodynamic results are presented in Table 2.6-7. Table 2.6-8 presents the distribution of capital costs for these points in both millions of dollars and in dollars per kilowatt of station dispatched power. Table 2.6-9 presents the makeup of net station

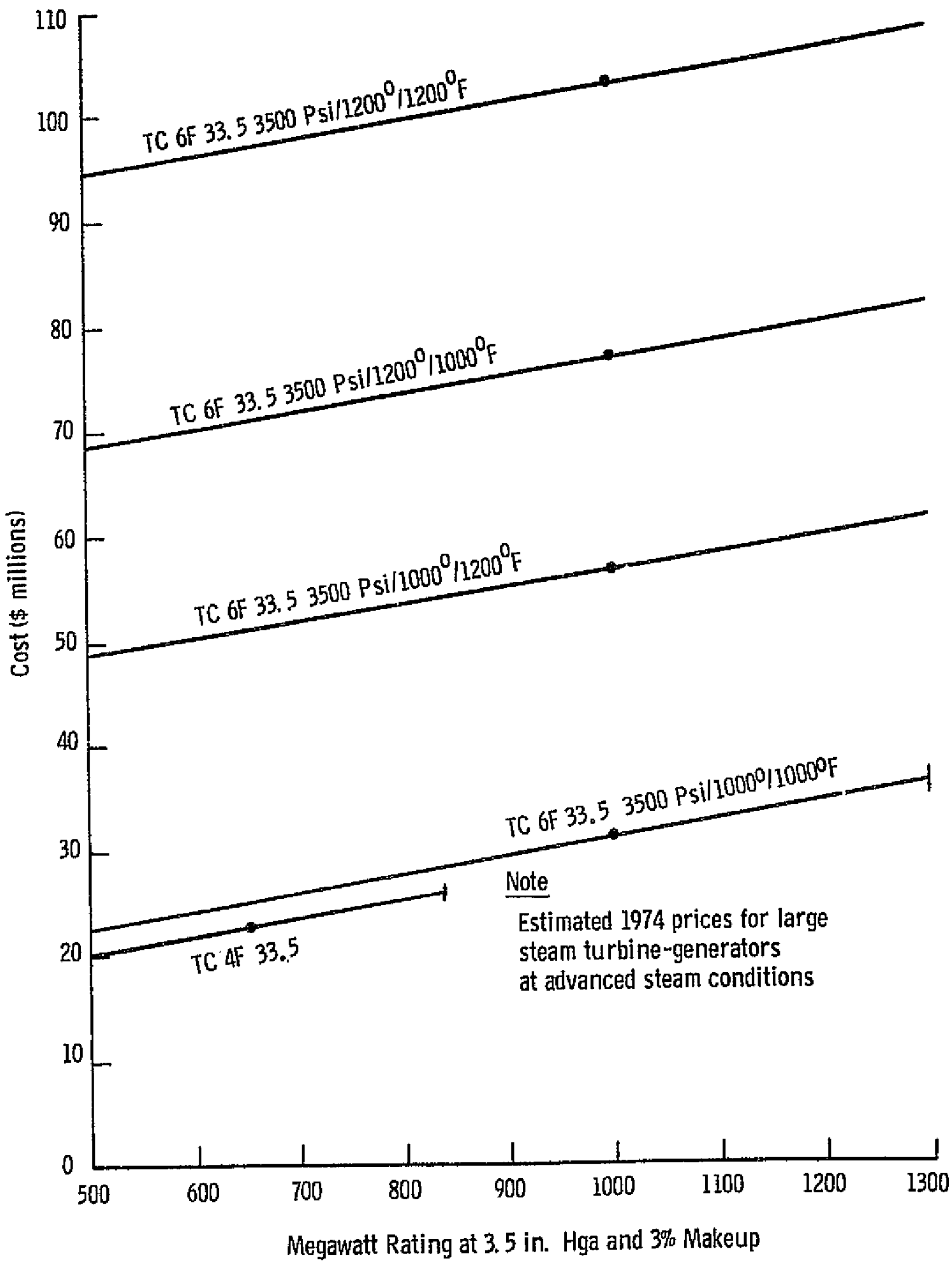


Figure 2.6-8. Cost Basis-Advanced Steam

dispatched power from the power generation from steam and gas turbine components reduced by the several auxiliary demands and by the final transformer loss.

The interplay between these cases will be made apparent by discussion of particular clusters of related cases.

FOLDOUT FRAME /

Parameters	Case 1*	2	3	4	5
Power Output (MWe)	745	559	1119	1484	745
Furnace, Coal, and Conversion Process	AFB Ill. #6				
Prime Cycle					
Throttle					
Psig	3500				
°F	1200				
Reheat (°F)	1000				
Second reheat (°F)	--	—	--	--	--
Feed (°F)	510				560
Condense (Hga)	1.5				
Combustion Air Supply					
Excess air (percent)	--	--	--	--	--
Pressure ratio	--	--	--	--	--
Turbine inlet temperature (°F)	--	--	--	--	--
Exhaust energy	--	--	--	--	--
Actual Powerplant Output (MWe)	745	559	1119	1484	745
Thermodynamic Efficiency (percent)	45.4	46.1	45.4	45.4	45.7
Powerplant Efficiency (percent)	37.7	38.3	37.8	37.6	38.0
Overall Energy Efficiency (percent)	37.7	38.3	37.8	37.6	38.0
Coal Consumption (lb/kWh)	0.84	0.83	0.84	0.84	0.83
Plant Capital Cost ($ million)	538	437	824	1077	539
Plant Capital Cost ($/kWe)	722	782	736	725	723
Cost of Electricity, Capacity Factor = 0.65					
Capital (mills/kWh)	22.8	24.7	23.3	22.9	22.9
Fuel (mills/kWh)	7.7	7.6	7.7	7.7	7.6
Maintenance and operating (mills/kWh)	2.5	2.7	2.3	2.2	2.5
Total (mills/kWh)	33.1	35.0	33.3	32.8	33.0
Sensitivity					
Capacity factor = 0.50 (total mills/kWh)	40.7	43.2	41.0	40.4	40.6
Capacity factor = 0.80 (total mills/kWh)	28.3	29.9	28.5	28.1	28.3
Capital Δ = 20 percent (Δmills/kWh)	4.6	4.9	4.7	4.6	4.6
Fuel Δ = 20 percent (Δmills/kWh)	1.5	1.5	1.5	1.5	1.5
Estimated Time for Construction (years)	5	5	6	5	5
Estimated Date of 1st Commercial Service (year)	1987	1987	1987	1987	1987

*Base case.
**Dry cooling tower.

AFB = Atmospheric fluidized bed
CF = Conventional furnace
FW = Feedwater
Ill. = Illinois
LBtu = Low Btu
Mont = Montana
N. D. = North Dakota
PF = Pressurized Furnace
PFB
$(PFB)_R$
SRC

Table 2.6-7

ADVANCED STEAM CYCLE
PARAMETRIC VARIATIONS FOR TASK I STUDY

7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24	25	26	27	28
748	745	743	738	744	745	746	747	741	743	763	759	762	767	1716	1836	1799	745	782	741	1000	743
→	→	→	→	→	→	→	→	AFB N.D.	AFB Mont	CF Ill. #6	CF N.D.	CF Mont	CF SRC	PF Ill. #6 LBtu	PF N.D. LBtu	PF Mont LBtu	PFB Ill. #6	PFB N.D.	PFB Mont	(PFB)ᵣ Ill. #6	AFB Ill. #6
→	4000	3500	2400	3500 →	→	→	→	→	→	→	→	→	→	→	→	→	→	→	→	→	→
→	→	→	1000 →	→	→	→	→	1200 →	→	→	→	→	→	→	→	→	→	→	→	→	→
1400	1000 →	→	→	→	1200	1400	1200	1000 →	→	→	→	→	→	→	→	→	→	→	→	→	→
--	--	--	--	--	--	--	1200	--	--	--	--	--	--	--	--	--	--	--	--	--	--
→	→	→	480	510 →	→	→	547	510 →	→	→	→	→	→	→	→	→	→	→	→	→	→
→	→	3.45**	1.5 →	→	→	→	→	→	→	→	→	→	→	→	→	→	→	→	→	→	1.9**
			•																		
--	--	--	--	--	--	--	--	--	--	--	--	--	--	15 →	→	→	20 →	→	→	→	--
--	--	--	--	--	--	--	--	--	--	--	--	--	--	10 →	→	→	→	→	→	→	--
--	--	--	--	--	--	--	--	--	--	--	--	--	--	1800 →	→	→	1600 →	→	→	→	--
--	--	--	--	--	--	--	--	--	--	--	--	--	--	Steam for gasifier →	→	→	FW heat →	→	→	Regen 0.85	--
748	745	743	738	744	745	746	747	741	743	763	759	762	767	1716	1836	1799	745	782	741	1000	743
47.7	45.4	42.5	42.9	44.0	45.1	44.7	47.2	45.4	45.4	45.4	45.4	45.4	45.4	45.4	45.4	45.4	40.0	40.0	40.0	45.4	45.3
39.8	37.7	35.2	35.3	36.5	37.6	36.5	39.3	34.7	36.6	38.2	36.0	38.0	40.1	34.0	34.2	34.3	39.2	37.3	37.9	37.3	37.6
39.8	37.7	35.2	35.3	36.5	37.6	36.5	39.3	34.7	36.6	38.7	36.0	38.0	31.2	34.0	34.2	34.3	39.2	37.3	37.9	37.3	37.6
0.80	0.84	0.90	0.90	0.87	0.84	0.82	0.91	1.43	1.04	0.93	1.38	1.01	1.01	0.93	1.45	1.11	0.81	1.33	1.01	0.85	0.84
633	543	567	461	454	501	540	554	569	539	574	621	603	507	1224	1380	1295	561	609	571	682	608
847	728	763	623	610	672	735	742	767	725	778	817	792	660	713	751	719	752	778	771	681	818
26.8	23.0	24.1	19.7	19.3	21.3	23.3	23.5	24.3	22.9	24.6	25.8	25.0	20.9	22.5	23.8	22.8	23.8	24.6	24.4	21.5	25.9
7.3	7.7	8.2	8.2	7.9	7.7	7.5	7.4	8.4	7.9	7.6	8.1	7.6	15.3	8.5	8.5	8.5	7.4	7.8	7.7	7.8	7.7
7.5	2.5	2.6	2.6	2.5	2.5	2.5	2.5	2.6	2.5	2.2	2.3	2.7	2.4	3.3	3.1	3.4	3.4	3.3	3.4	2.9	2.5
36.6	33.3	34.9	30.5	29.8	31.5	33.3	33.3	35.2	33.4	34.4	36.2	34.9	38.6	34.3	35.3	34.6	34.6	35.7	35.4	32.2	36.1
45.4	40.9	43.0	37.2	36.3	38.6	41.0	41.1	43.3	41.9	42.5	44.6	43.1	45.6	42.1	43.4	42.4	42.8	44.1	43.8	39.6	44.7
31.1	28.5	29.9	26.3	25.7	27.1	28.5	29.5	30.2	28.6	29.4	30.9	29.8	34.3	29.5	30.3	29.7	29.5	30.5	30.2	27.7	30.8
5.4	4.6	4.8	3.9	3.9	4.3	4.7	4.7	4.9	4.6	4.9	5.2	5.0	4.2	4.5	4.8	4.6	4.8	4.9	4.9	4.3	5.2
1.5	1.5	1.6	1.6	1.6	1.5	1.5	1.5	1.7	1.6	1.5	1.6	1.5	3.1	1.7	1.7	1.7	1.5	1.6	1.5	1.6	1.5
5	5	5	5	5	5	5	5	5	5	5	5	5	5	5	5	5	5	5	5	5	5
1995	1987	1987	1985	1985	1987	1995	1990	1987	1987	1987	1987	1987	1987	1987	1987	1987	1987	1987	1987	1987	1987

ressurized fluidized bed
ressurized fluidized bed (recuperative)
olvent refined coal

Table 2.6-6

SUMMARY SHEET
ADVANCED STEAM CYCLE BASE CASE

CYCLE PARAMETER

Net Power Output (MWe)	745
Furnace and Coal Type	Atmospheric fluidized bed Illinois No. 6
Prime Cycle	
Throttle	
Psig	3500
°F	1200
Reheat (°F)	1000
Feed (°F)	510
Condenser (Hga)	1.5
Heat Rejection	Wet cooling tower

PERFORMANCE AND COST

Thermodynamic efficiency (percent)	45.4
Powerplant efficiency (percent)	37.7
Overall energy efficiency (percent)	37.7
Plant capital cost ($ x 10^6)	538
Plant capital cost ($/kWe)	722
Cost of electricity (mills/kWh)	33.1

[illegible]TURAL RESOURCES

Coal ([illegible]Wh)	0.84
Water (gal/kWh)	
Total	0.423
Cooling	0.393
Processing	0
Makeup	0.030
NO_x suppression	0
Stack gas cleanup	0
Land (acres/100 MWe)	4.70

ENVIRONMENTAL INTRUSION

	Lb/10^6-Btu Input	Lb/kWh Output
SO_2	1.03	9.79×10^{-3}
NO_x	0.27	2.45×10^{-3}
HC	--	--
CO	0.21	1.90×10^{-3}
Particulates	0.10	0.91×10^{-3}

	Btu/kWh
Heat to water	4369
Heat, total rejected wastes	5638

	Lb/kWh	Lb/Day
Wastes		
Furnace solids	0.146	2.615×10^6
Fine dust from cyclones	0.113	2.022×10^6
Fly ash	0.016	0.283×10^6

MAJOR COMPONENT CHARACTERISTICS

Major Component	Unit or Module: Size (ft) (W x L (or D) x H)	Unit or Module: Weight (lb) (x 10^6)	Unit or Module: Cost ($ x 10^6)	Units Required	Total Cost ($ x 10^6)	$/kW Output
Steam turbine-generator	30 x 174 x 25	6.5	69.4	1	69.4	93.1
Furnace modules	12 x 30 x 150	2.4	10.3	3.89	40.2	54.0

Table 2.6-8 (Page 1 of 3)

CAPITAL COSTS DISTRIBUTIONS FOR ADVANCED STEAM CYCLE

	CASE NO.	1	2	3	4	5	6	7	8	9	10
MAJOR COMPONENTS											
PRIME CYCLE											
STEAM TURB-GEN	MM$	69.4	66.6	79.4	138.8	69.4	95.4	121.4	70.2	69.7	24.5
PRIMARY HEAT INPUT AND FUEL SYSTEM											
FURNACE MODULES	MM$	40.2	29.7	60.3	80.4	40.6	40.2	40.5	41.9	43.0	42.2
LOW TEMP AIR PREHEATER	MM$	2.8	2.1	4.3	5.7	2.8	2.8	2.7	2.8	3.0	3.0
PRESSURIZING GAS TURBINE (COMP-GEN-HEAT EXCH)	MM$	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.
GASIFIER (INCLUDING BOOST STEAM TURB-COMP	MM$	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.
SUB-TOTAL OF MAJOR COMPONENTS	MM$	112.4	98.4	143.9	224.8	112.8	138.3	164.6	114.9	115.7	69.7
BALANCE OF PLANT											
COOLING TOWER	MM$	5.0	3.8	7.5	10.0	5.0	5.0	5.0	5.0	13.5	5.0
STACK-GAS CLEAN-UP EQUIP.	MM$	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.
ALL OTHER	MM$	141.1	109.2	211.7	282.3	141.1	141.1	141.1	141.1	138.9	141.1
SITE LABOR	MM$	37.2	28.7	55.8	74.4	37.2	37.2	37.2	37.2	43.6	37.2
SUB-TOTAL OF BALANCE OF PLANT	MM$	183.3	141.7	275.0	366.7	183.3	183.3	183.3	183.3	196.0	183.3
CONTINGENCY	MM$	59.2	48.0	83.8	118.3	59.2	64.3	69.6	59.6	62.3	50.6
ESCALATION COSTS	MM$	84.8	68.8	143.8	169.7	84.9	92.3	99.8	85.5	89.4	72.6
INTEREST DURING CONSTRUCTION	MM$	98.7	80.1	177.1	197.4	98.8	107.3	116.1	99.5	104.0	84.4
TOTAL CAPITAL COST	MM$	538.4	437.0	823.6	1076.8	539.2	585.6	633.4	542.9	567.4	460.6
MAJOR COMPONENTS COST	$/KWE	150.9	176.0	128.7	151.5	151.4	185.4	220.2	154.2	155.7	94.4
BALANCE OF PLANT	$/KWE	246.1	253.5	245.8	247.0	246.0	245.7	245.2	246.1	263.8	248.3
CONTINGENCY	$/KWE	79.4	85.9	74.9	79.7	79.5	86.2	93.1	80.1	83.9	68.5
ESCALATION COSTS	$/KWE	113.8	123.2	128.5	114.3	114.0	123.6	133.5	114.8	120.3	98.3
INTEREST DURING CONSTRUCTION	$/KWE	132.4	143.3	158.3	133.0	132.6	143.8	155.3	133.6	140.0	114.3
TOTAL CAPITAL COST	$/KWE	722.6	782.0	736.3	725.5	723.4	784.8	847.2	728.7	763.8	623.9

Table 2.6-8 (Page 2 of 3)

CAPITAL COSTS DISTRIBUTIONS FOR ADVANCED STEAM CYCLE

	CASE NO.	11	12	13	14	15	16	17	18	19	20
MAJOR COMPONENTS											
PRIME CYCLE											
STEAM TURB-GEN	MM$	23.8	49.4	74.9	80.8	69.4	69.4	69.4	69.4	69.4	69.4
PRIMARY HEAT INPUT AND FUEL SYSTEM											
FURNACE MODULES	MM$	39.3	39.5	40.2	37.7	45.6	40.5	32.1	36.6	33.6	28.7
LOW TEMP AIR PREHEATER	MM$	2.9	2.9	2.8	2.7	3.2	3.0	2.5	2.6	2.4	2.3
PRESSURIZING GAS TURBINE (COMP-GEN-HEAT EXCH)	MM$	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.
GASIFIER (INCLUDING BOOST STEAM TURB-COMP	MM$	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.
SUB-TOTAL OF MAJOR COMPONENTS	MM$	66.0	91.7	118.0	121.2	118.2	112.8	104.0	108.5	105.4	100.4
BALANCE OF PLANT											
COOLING TOWER	MM$	5.0	5.0	5.0	5.0	5.0	5.0	5.0	5.0	5.0	5.0
STACK-GAS CLEAN-UP EQUIP.	MM$	0.	0.	0.	0.	0.	0.	32.4	32.4	32.4	13.0
ALL OTHER	MM$	141.1	141.1	141.1	141.1	146.2	141.1	145.3	152.3	147.9	130.7
SITE LABOR	MM$	37.2	37 [illegible]	37.2	37.2	43.0	37.2	39.7	42.7	40.8	29.5
SUB-TOTAL OF BALANCE OF PLANT	MM$	183.3	183. [illegible]	183.3	183.3	194.2	183.3	222.4	232.4	226.1	178.2
CONTINGENCY	MM$	49.9	55.0	60.3	60.9	62.5	59.2	65.3	68.2	66.3	55.7
ESCALATION COSTS	MM$	71.5	78.9	86.4	87.3	89.6	84.9	93.6	97.8	95.1	79.9
INTEREST DURING CONSTRUCTION	MM$	83.2	91.8	100.5	101.6	104.2	98.8	108.9	113.7	110.6	92.9
TOTAL CAPITAL COST	MM$	454.0	500.8	548.5	554.4	568.6	539.2	594.1	620.6	603.4	507.1
MAJOR COMPONENTS COST	$/KWE	88.7	123.2	158.2	162.3	159.4	151.8	136.3	142.9	138.3	130.8
BALANCE OF PLANT	$/KWE	246.4	246.1	245.9	245.5	262.0	246.7	291.6	306.2	296.8	232.2
CONTINGENCY	$/KWE	67.0	73.9	80.8	81.6	84.3	79.7	85.6	89.8	87.0	72.6
ESCALATION COSTS	$/KWE	96.1	105.9	115.9	117.0	120.9	114.3	122.7	128.8	124.8	104.1
INTEREST DURING CONSTRUCTION	$/KWE	111.8	123.2	134.8	136.1	140.6	133.0	142.8	149.8	145.2	121.1
TOTAL CAPITAL COST	$/KWE	610.1	672.2	735.6	742.4	767.2	725.5	778.9	817.5	792.1	660.8

Table 2.6-8 (Page 3 of 3)

CAPITAL COSTS DISTRIBUTIONS FOR ADVANCED STEAM CYCLE

	CASE NO.	21	22	23	24	25	26	27	28
MAJOR COMPONENTS									
PRIME CYCLE									
STEAM TURB-GEN	MM$	69.4	69.4	69.4	66.6	66.6	66.6	69.4	69.4
PRIMARY HEAT INPUT AND FUEL SYSTEM									
FURNACE MODULES	MM$	11.2	11.4	11.2	53.2	58.6	52.8	79.5	40.3
LOW TEMP AIR PREHEATER	MM$	0.	0.	0.	0.	0.	0.	0.	2.8
PRESSURIZING GAS TURBINE (COMP-GEN-HEAT EXCH)	MM$	90.2	98.7	95.4	21.0	22.9	21.4	39.4	0.
GASIFIER (INCLUDING BOOST STEAM TURB-COMP	MM$	294.2	356.2	320.1	0.	0.	0.	0.	0.
SUB-TOTAL OF MAJOR COMPONENTS	MM$	464.9	535.7	496.1	140.8	148.1	140.7	188.2	112.9
BALANCE OF PLANT									
COOLING TOWER	MM$	6.0	6.0	6.0	6.0	6.0	6.0	6.0	20.0
STACK-GAS CLEAN-UP EQUIP.	MM$	0.	0.	0.	0.	0.	0.	0.	0.
ALL OTHER	MM$	154.7	164.8	160.0	130.2	144.0	135.7	142.5	153.2
SITE LABOR	MM$	46.8	51.4	49.3	31.1	36.3	31.5	37.8	48.5
SUB-TOTAL OF BALANCE OF PLANT	MM$	207.5	222.2	215.3	167.3	186.3	173.2	186.3	221.7
CONTINGENCY	MM$	134.5	151.6	142.3	61.6	66.9	62.8	74.9	66.8
ESCALATION COSTS	MM$	192.9	217.4	204.0	88.4	95.9	90.0	107.4	95.9
INTEREST DURING CONSTRUCTION	MM$	224.4	252.9	237.4	102.8	111.6	104.7	125.0	111.5
TOTAL CAPITAL COST	MM$	1224.1	1379.8	1295.1	560.9	608.7	571.4	681.9	608.4
MAJOR COMPONENTS COST	$/KWE	270.9	291.7	275.8	189.0	189.5	189.9	188.2	151.4
BALANCE OF PLANT	$/KWE	120.9	121.0	119.7	224.6	238.3	233.7	186.3	298.3
CONTINGENCY	$/KWE	78.4	82.5	79.1	82.7	85.5	84.7	74.9	89.9
ESCALATION COSTS	$/KWE	112.4	118.4	113.4	118.6	122.7	121.5	107.4	129.0
INTEREST DURING CONSTRUCTION	$/KWE	130.7	137.7	132.0	138.0	142.7	141.4	124.9	150.0
TOTAL CAPITAL COST	$/KWE	713.2	751.4	720.0	752.8	778.7	771.2	681.6	818.6

Table 2.6-9

POWER OUTPUT AND AUXILIARY POWER DEMAND FOR BASE CASE AND PARAMETRIC VARIATIONS: ADVANCED STEAM CYCLE

CASE NO.		1	2	3	4	5	6	7	8	9	10
PRIME CYCLE POWER OUTPUT	MW	800.0	600.0	1200.0	1600.0	800.0	800.0	800.0	800.0	800.0	800.0
BOTTOMING CYCLE POWER OUTPUT	MW	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.
FURNACE POWER OUTPUT	MW	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.
BALANCE OF PLANT AUX. POWER REQ'D.	MW	19.0	14.6	27.5	44.0	19.0	18.7	[illegible]	19.0	19.0	19.0
FURNACE AUX. POWER REQ'D.	MW	31.9	23.6	47.9	63.8	31.7	31.2	30.[illegible]	31.9	34.1	38.7
TRANSFORMER LOSSES	MW	4.0	3.0	6.0	8.0	4.0	4.0	4.0	4.0	4.0	4.0
NET STATION OUTPUT	MW	745.1	558.7	1118.6	1484.2	745.3	746.2	747.7	745.1	742.9	738.3

CASE NO.		11	12	13	14	15	16	17	18	19	20
PRIME CYCLE POWER OUTPUT	MW	800.0	800.0	800.0	800.0	800.0	800.0	800.0	800.0	800.0	800.0
BOTTOMING CYCLE POWER OUTPUT	MW	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.
FURNACE POWER OUTPUT	MW	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.
BALANCE OF PLANT AUX. POWER REQ'D.	MW	19.0	19.0	19.0	18.5	19.0	19.0	20.0	20.0	20.0	20.0
FURNACE AUX. POWER REQ'D.	MW	32.9	32.1	31.3	30.7	35.9	33.8	13.2	16.9	14.2	8.6
TRANSFORMER LOSSES	MW	4.0	4.0	4.0	4.0	4.0	4.0	4.0	4.0	4.0	4.0
NET STATION OUTPUT	MW	744.1	744.9	745.7	746.8	741.1	743.2	762.8	759.1	761.8	767.4

CASE NO.		21	22	23	24	25	26	27	28
PRIME CYCLE POWER OUTPUT	MW	800.0	800.0	800.0	600.0	600.0	600.0	800.0	800.0
BOTTOMING CYCLE POWER OUTPUT	MW	0.	0.	0.	0.	0.	0.	0.	0.
FURNACE POWER OUTPUT	MW	944.0	1064.6	1027.0	177.0	213.8	171.4	237.0	0.
BALANCE OF PLANT AUX. POWER REQ'D.	MW	19.0	19.0	19.0	19.0	19.0	19.0	19.0	20.9
FURNACE AUX. POWER REQ'D.	MW	0.	0.	0.	9.0	9.0	7.6	12.4	31.9
TRANSFORMER LOSSES	MW	8.7	9.3	9.1	3.9	4.1	3.9	5.2	4.0
NET STATION OUTPUT	MW	1716.3	1836.3	1798.9	745.1	781.7	740.9	1000.4	743.2

DISCUSSION OF RESULTS

Plant Size Influence

It was anticipated that some advantage would be gained through increased plant size. Cases 1 through 4 explore this at generator ratings of 800, 600, 1200, and 1600 MW, respectively. The results show that the 600 MW plant, Case 2, is indeed more costly, but all the others show little differentiation. The balance of plant costs were nearly identical at $246/kW. The modular nature of the furnaces resulted also in a uniform cost per kilowatt. The steam turbine-generators showed distinct differences. The 600 MW unit was most costly per kilowatt. The 1600 MW plant was composed of two 800 MW units, so it was identical to the base case for 800 MW. The 1200 MW unit using a six-flow turbine was the least expensive turbine-generator. This saving was offset by the added year of construction time attributed to this very large unit. The result was that above 600 MW there was a nearly constant capital cost resulting in a per unit generation cost of 33 mills/kWh for the advanced steam turbine at conditions of 3500 psig, 1200 F/1000 F (2.42×10^7 N/m^2, 922 K/811 K) using the atmospheric fluidized bed steam generators.

Advanced Steam Condition Influences

Ten variations were explored about the base case conditions with nominal 800 MW generation (738 to 748 MW net station output) using atmospheric fluidized bed steam generators and Illinois No. 6 coal. In every case the balance of plant cost was between 245 and 248 dollars per kilowatt. Except for the two cases noted the throttle pressure was 3500 psig (2.42×10^7 N/m^2). Table 2.6-10 presents the thermodynamic and economic results of these evaluations in the order of increasing overall energy efficiency.

These results clearly show that the fuel savings resulting from increased efficiency do not offset the increased cost of major equipment. The two cases with conventional 1000 F/1000 F (811 K/811 K) throttle and reheat are distinctly more economic then the higher temperature cases. Case 12 with conventional throttle conditions and 1200 F (922 K) reheat is the economically superior case with advanced steam conditions, and it realizes a one point efficiency advantage over standard conditions. This particular configuration avoids advanced temperature in the supercritical pressure sections of the steam generator and the steam turbine and throttle valves. The high temperature is realized at a more modest level of 700 psi (4.82×10^6 N/m^2). These several factors result in the recommendation that these conditions be considered for Task II. The heater above reheat point (HARP) variation of Case 5 shows both modest thermodynamic and economic advantages over the base case. The double reheat Case 14 should be compared with Case 12. The result is that a second reheat achieves measurable thermodynamic gain, but at the expense of a 6 percent poorer cost of electricity.

Table 2.6-10

THERMODYNAMIC AND ECONOMIC IMPACT OF
ADVANCED TURBINE STEAM CONDITIONS

Throttle and Reheat Conditions (psig/°F/°F)	Overall Energy Efficiency (%)	Electricity Cost (mills/kWh)	Case Number
2400/1000/1000	35.3	30.5	10
3500/1000/1000	36.5	29.8	11
3500/1000/1200	37.5	31.5	12
3500/1200/1000 (Base Case)	37.7	33.1	1
4000/1200/1000	37.7	33.3	8
3500/1200/1000 560 F HARP*	38.0	33.0	5
3500/1000/1400	38.5	33.3	13
3500/1200/1200	38.7	34.8	6
3500/1000/1200/1200 F	39.3	33.3	14
3500/1200/1400	39.8	36.6	7

*HARP = heater above reheat point

Fuel and Combustion Option Influences

The effects due to use of various fuels were investigated through thirteen parametric points wherein the steam turbine conditions were those of the base case, 3500 psig, 1200 F/1000 F (2.42×10^7 N/m^2, 922 K/811 K), condensing at 1.5 inches mercury absolute (5.07×10^3 N/m^2). In all cases the progression of electric costs was least for Illinois No. 6 coal, intermediate for Montana Sub-Bituminous coal, and greatest for North Dakota Lignite, with the single case with Solvent Refined Coal showing the highest electric cost. These points also show that among the various combustion options the progression was as follows: least costly—atmospheric fluidized bed; pressure-fired low-Btu integrated plant second; conventional fired plant third; and the pressurized fluidized bed plant most expensive. However, Case 27 with an 800 MW steam turbine-generator in a pressurized fluidized bed plant produced less costly electricity than any of the foregoing cases. This indicates that the three PFB cases may have been adversely affected by the choice of a 600 MW steam turbine for their evaluation.

The comparison of electricity production costs for the thirteen cases is presented in Table 2.6-11.

Table 2.6-11

COMPARISON OF FUEL AND COMBUSTION EFFECTS ON ELECTRIC PRODUCTION COSTS, Mills/kWh (Case No.)

Combustion Configuration	Ill.No.6	N.D.	Mont.	SRC
Pressurized fluidized bed (recuperative)	32.2(27)	—	—	—
Atmospheric fluidized bed	33.1(1)	35.2(15)	33.4(16)	—
Pressure-fired low-Btu-combined	34.3(21)	35.3(22)	34.6(23)	—
Conventional fired boiler	34.4(17)	36.2(18)	34.9(19)	38.6(20)
Pressurized fluidized bed (feedwater heating)	34.6(24)	35.7(25)	35.4(26)	—

Note: Ill. = Illinois; N.D. = North Dakota; Mont. = Montana; SRC = solvent refined coal

Dry Cooling Tower Influences

Cases 9 and 28 as contrasted to Base Case 1 show an increase of 1.8 mills/kWh for use of 3.45 in. Hga (1.16×10^4 N/m^2) with a 60 F (288 K) initial temperature difference dry cooling tower, and an increase of 3.0 mills/kWh for use of 1.9 in. Hga (6.42×10^3 N/m^2) with a 40 F (277 K) initial temperature difference dry cooling tower.

Observations

The capital cost distributions are presented in Table 2.6-8 with a summary in Table 2.6-12. Because of the modular nature of the AFB and PFB steam generators, the major variation in major component costs are due to advanced steam turbines or addition of gas turbines. The balance of plant costs are fairly uniform in dollars per kilowatt for each type of configuration. The low value for the pressure-fired low-Btu gas Cases 21 thru 23 correlates with the gasifier and bottoming steam turbine cost allocation to major equipment. The low value for balance of plant for the single PFB case with an 800 MW steam turbine using recuperative gas turbines, Case 27, indicates a 10 percent increase in balance of plant for a 33 percent increase in rating for most elements of the plant as compared with the three earlier PFB Cases 24 thru 26. The AFB plant with conventional 3500 psig, 1000 F/1000 F (2.42×10^7 N/m^2, 811 K/811 K) conditions, Case 11,

shows that the capital advantage accrues almost entirely to reduced steam turbine-generator cost and not to the balance of plant nor to furnace module cost as compared with Base Case 1.

Table 2.6-12

CAPITAL COST SUMMARY FOR STEAM PLANTS OF 3500 PSIG, 1200 F/1000 F

Configuration	AFB	AFB*	CF	PFB	$(PFB)_R$	PF
Major components $/kWe	150	90	140	190	190	270
Balance of plant $/kWe	250	250	290	225	190	120
Total $/kWe	725	610	780	750	680	715
Case no.	1	11	17	24	27	21

*3500 psig/1000 F/1000 F steam conditions

Note: AFB = atmospheric fluidized bed
CF = conventional furnace
PF = pressure-fired
PFB = pressurized fluidized bed
$(PFB)_R$ = pressurized fluidized bed, recuperative

Dominant cost factors for all plants were the balance of plant, contingency, and interest during construction. The pressure-fired boilers presented an unusually complex sequence of apparatus that would merit greater efforts toward integration and simplification. For example, the steam produced by the heat recovery steam generators that follow the gas turbines would be more efficiencly expanded in the large main turbine rather than in a separate bottoming steam turbine.

RECOMMENDED CASES

The atmospheric fluidized bed furnace-steam generator with 3500 psig, 1000 F (2.42×10^7 N/m^2, 811 K), throttle and reheat to 1200 F (922 K) (Case 12) is recommended for further study in Task II. Table 2.6-10 shows that this case increases efficiency at a minimal increase in cost of electricity as compared with conventional steam conditions for Case 11.

In addition each of the following cycle modifications though not studied would contribute to increased efficiency or reduced cost and may be more beneficial than the substantial departures from state-of-the-art temperatures that were investigated.

1. Heater above the reheat point (HARP)
2. Throttle temperature at 1050 F (839 K)

3. Unit rating to utilize maximum limiting flow to last stages of turbine

4. Condenser pressure optimized for wet cooling tower in the range of 2.0 to 3.5 inches of mercury absolute pressure (6.75×10^3 to 1.18×10^4 N/m^2)

5. To achieve the 37.5 percent overall efficiency of the 3500 psig, 1000 F/1200 F (2.42×10^7 N/m^2, 811 K/922 K) recommended Case 12, utilize a combination of state of the art conditions of 4000 psig, 1000 F throttle, 1025 F first reheat, and 1050 F second reheat (2.77×10^7 N/m^2, 811 K/825 K/839 K).

REFERENCES

1. Spencer, R.C., Cotton, K.C., and Cannon, C.N., "A Method for Predicting the Performance of Steam Turbine-Generators... 16,500 kW and Larger," ASME Paper 62-WA-209, November 1962.

2. "Heat Rates for Fossil Reheat Cycles Using General Electric Steam Turbine-Generators 150,000 kW and Larger," GET-2050C, February 1974.

2.7 LIQUID METAL TOPPING CYCLE

DESCRIPTION OF CYCLE

The liquid metal topping cycle is described in Figure 2.7-1, which shows the arrangement of components for the potassium base case (Case 1). The arrangement for the cesium base case (Case 17) is identical to Case 1. The parameters corresponding to each of the 18 liquid metal topping cases are shown later in this section under "Discussion of Results."

The system shown in Figure 2.7-1 burns coal directly in atmospheric fluidized bed furnaces, in which heat is transferred to potassium at a boiling temperature of 1400 F (1033 K). The potassium vapor, at a saturated condition, enters six double-flow turbines. After expansion through the turbines, the wet vapor is condensed, and the rejected heat is used to boil and reheat steam for the bottoming cycle. A set of liquid metal pumps returns liquid potassium to the boiler.

The bottoming cycle is a steam system with conventional temperatures of 1000 F (811 K) leaving the boiler and reheater, and a pressure of 3515 psia (24.2 MN/m^2) leaving the boiler. Steam is condensed in a set of wet cooling towers.

Major variations of components from the base cases included substitution of pressurized furnaces (PF) and pressurized fluidized beds (PFB) for the atmospheric fluidized bed (AFB) furnaces, and substitution of dry cooling towers for the wet cooling towers. One case also considered the addition of a regenerative feed heater in the liquid metal circuit.

Most of the cases were run with potassium, and only two cases used cesium as the topping cycle working fluid. Potassium was favored because substantially more information exists for component performance and material compatibility for potassium than for cesium.

A turbine inlet temperature of 1400 F (1033 K) was selected for most cases because considerable turbine testing has been done in this temperature range, and less expensive alloys can be selected for operation at this lower temperature.

The atmospheric fluidized bed is limited in temperature to about 1550 F (1116 K) for reasons of degradation of sulfur removal at higher temperatures. Therefore, the potassium turbine inlet temperature is limited to about 1500 F or below with the atmospheric fluidized bed. A higher turbine inlet temperature was investigated with a pressurized furnace.

The condensing temperature of potassium was set at 1100 F (866 K). At lower temperatures the vapor density decreases considerably, thus greatly enlarging the low-pressure turbine stages.

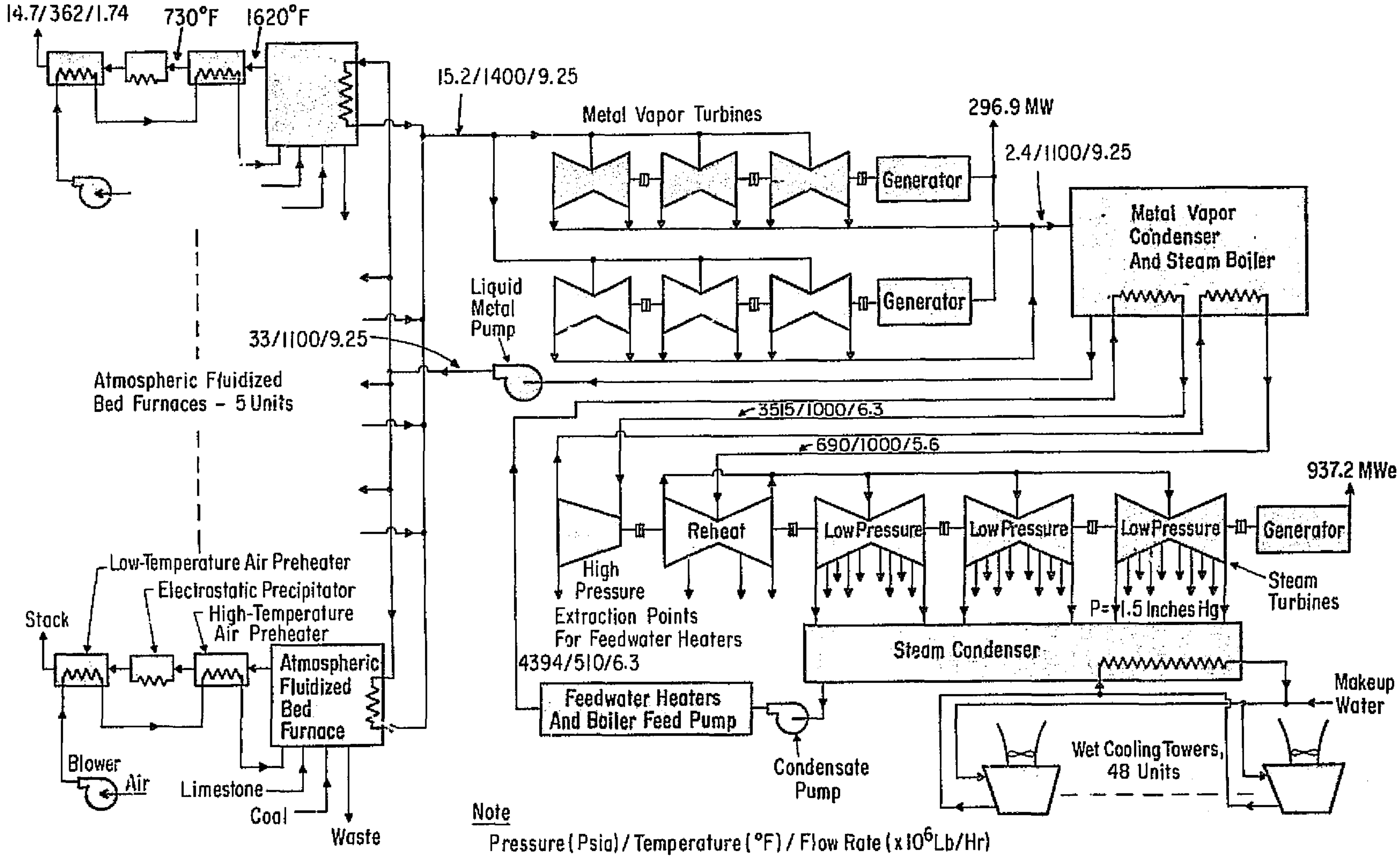


Figure 2.7-1. Liquid Metal Topping Cycle

Because this condition is not so severe with cesium, the condensing temperature for the cesium base case was lowered to 1000 F (811 K). A second cesium case was run at 1100 F condensing temperature, for comparison with the potassium base case.

ANALYTICAL PROCEDURE AND ASSUMPTIONS

The starting point for the analysis of liquid metal topping systems was a report (ref. 1) covering earlier work on potassium topping systems for central station power. That report gives details on the basic analytical procedures and assumptions, and a list of references to the literature that served as background for the present study.

Cycle Calculations

Cycle calculations were carried out with the use of a computer program described in Reference 1. The computer program calculated the entire power system, including the furnace and the bottoming cycle. However, only results from the liquid metal part of the calculation were used in this study; the furnace/liquid-metal boiler, bottoming cycle, etc., were calculated separately using the same procedures that were applied to all other energy conversion systems in this study.

Inputs to the liquid metal calculation included liquid metal turbine efficiency, liquid metal pump efficiency, and pressures and temperatures throughout the liquid metal circuit.

Properties of potassium and cesium were tabulated from References 2 and 3, and were included as input data to the computer calculation. Power output from the steam bottoming system was an input to the calculation. From this input, flow rates in the steam bottoming system were computed, from which liquid metal flow rates and power output from the liquid metal topping system were calculated. Liquid metal pump power was subtracted from the generator power output to arrive at a net power output from the liquid metal topping system.

Metal Vapor Turbine

A separate computer program was used for calculation of the metal vapor turbine. The performance of each stage was calculated sequentially, using the Ainley and Mathieson method (ref. 4), which is a one-dimensional pitchline analysis. This analysis accounts for profile losses, secondary flow losses, and tip clearance losses.

Free vortex flow was assumed in calculating the hub-to-tip velocity distribution. This distribution was obtained to check whether there was negative reaction at the hub. In cases where negative reaction was found, corrective design changes were made.

The calculation method was verified for small-scale turbines by comparing predicted performance with actual performance of small turbines developed for space power systems.

Turbines were designed with as many as four stages. For the high-temperature cases (Cases 10 and 11), separate high-pressure turbines were designed with a single stage. In most cases, the turbines were designed in modules with a double-flow arrangement, that is, with each module having two sets of turbine stages in parallel.

Turbine speeds ranged from 1200 to 3600 RPM. Blade root diameters were designed within the range of 60 to 80 in. (1.52 to 2.03 m).

A turbine efficiency of 81 percent was calculated for the base case, and this isentropic efficiency was applied to all cases in this study.

Further design details of the metal vapor turbines can be found in Reference 1.

Liquid Metal Pump

Liquid metal pumps are centrifugal pumps of the type manufactured for the liquid metal fast breeder reactor program. Each pump has a capacity of 4500 gal/min (0.284 m^3/sec)

Liquid Metal Dump Tanks

The liquid metal dump tanks are for storing potassium or cesium while the system is not operating. The potassium or cesium is maintained in the liquid state by circulating hot gas in the dump tank jacket.

The dump tank also serves to maintain cleanliness of the liquid metal by means of a zirconium getter within the tank.

Stress

Stress calculations were performed only on the rotor of the metal vapor turbine. Stresses were not calculated in walls of liquid metal piping and vessels because low pressures resulted in low stresses in those regions.

Within the metal vapor turbine, stresses were calculated at the bucket roots, and in the high stress region near the center of the unbored wheel. The maximum allowable stress was based on a creep criterion of 0.2 percent over a thirty-year life (at a capacity factor of 65 percent) at operating temperature.

Transient thermal stresses were not calculated.

DESIGN AND COST BASIS

Size, weight, and cost of the major components of the liquid metal circuit were estimated, using the results of Reference 1 as a starting point. In this section, the derivations of these sizes, weights, and costs will be described.

Since the costs of Reference 1 were in terms of 1972 dollars, all costs were escalated by 8 percent the first year, and 10 percent the second year, to put the costs on the basis of 1974 dollars. The cost multiplying factor was 1.08 x 1.10, or 1.188.

Metal Vapor Turbine

The designs of potassium turbines are described in Reference 1 (sections beginning on pp. 21, 24, and 49). The sizes, weights, and costs for the present study were derived from those designs, using conventional scaling laws that apply to turbomachinery.

For turbomachinery, weight varies directly with volume, and power varies directly with flow or annulus area, assuming geometric similarity. Thus, weight = KL^3, and power = CL^2, where K and C are constants, and L is a characteristic length, for example, the length of the last stage blade. The weight per unit power is therefore approximately proportional to the 3/2 power of L. These relationships were used to scale sizes, weights, and costs from the original designs to the present study.

Cost estimates for metal vapor turbines were made in Reference 1, and summaries are given (pp. 65, 68-71).

Turbine sizes, weights, and costs for the various cases are summarized below.

Cases 1 through 9, and 12 through 16. For these cases, about 300 MW are to be generated by the potassium turbines out of a nominal plant total of 1200 MW. Since turbine temperatures and pressures are the same for all these cases, a single turbine design was used. The basic design used was the four-stage turbine described in Figure 13 and Table 16 of Reference 1 (pp. 52, 53). That turbine was designed to produce 113 MW at 1200 RPM, using the same potassium conditions as the base case. By scaling this turbine to 2/3 size at 1800 RPM, the disk size is reduced from 120 in. to 80 in. (3.05 to 2.03 m) diameter and the turbine can produce $(2/3)^2$ x 113 or 50 MWe. Therefore six double-flow turbines the size of the original design would produce the desired 300 MWe.

After the stresses in the rotating parts were calculated, it was concluded that some of the disk materials were marginal for

the life. The second-stage disk was changed from René 41 to Astroloy, the third-stage disk was changed from Inco 706 to René 41, and the fourth-stage disk was changed from Inco 706 to Inco 718.

The costs were estimated as described in Table 24 of Reference 1 (p. 71), substituting materials as indicated above. The weight and cost of materials were scaled for the higher rotative speed and smaller size, using the scaling relationships described above. Labor costs were estimated as a function of last stage bucket height.

In summary, Figure 13 of Reference 1 is scaled by 2/3 to get the turbine dimensions at 1800 RPM. For the base case, six double-flow turbines driving two generators are required for 300 MW output. Each double-flow turbine module is 27 ft long and 15 ft high (8.23 and 4.57 m) (not including the generator). The total weight for six turbines is estimated as 2.22×10^6 lb (1.01×10^6 kg). The total cost of the six turbines with two generators is estimated to be $43.3 million.

Case 8, with a nominal capacity 50 percent greater than the rest of the cases, utilizes nine double-flow turbine modules of the same size as above, and three generators, for a total cost 50 percent greater than the rest of the cases.

Case 10. The turbine for Case 10 has an inlet temperature of 1700 F (1200 K)—the highest temperature studied. A potassium flow rate of 5.16 million lb per hour (39,000 kg/sec) is required, and a net output power from topping cycle of 283 MW is generated. A design similar to the high pressure turbine described in Table 10 of Reference 1 (p. 25) could be made, but the turbine would be only a single-flow machine. It was considered desirable to reduce the disk diameter from 120 in. to 60 in. (3.04 to 1.52 m) by increasing the rotative speed from 1800 to 3600 RPM and using three or four smaller turbines. Preliminary turbine design calculations indicated that three turbines with 60 in. diameter disks at 3600 RPM would be a feasible design. The stresses were calculated for the disk and blades, and the materials of Figure 3 of Reference 1 (p. 23) were considered satisfactory if a stress criterion of 1 percent creep in thirty years was assumed (instead of the normal 0.2 percent criterion). This increase in creep was considered necessary at the higher temperature.

The low-pressure turbine is similar to the turbine for the base case, but with slightly smaller flow area. The stresses were checked for the first stage which operates with 1450 F (1061 K) vapor and were found to be acceptable.

Condensed liquid metal is removed between the high-pressure and low-pressure turbines.

To summarize, there are two high-pressure turbines, one double-flow unit 16 ft long by 9 ft high (4.9 by 2.7 m), and one single-flow unit 11 ft long and 9 ft high (3.35 by 2.74 m) at 3600 RPM which generate 112 MW. The weight estimate is 157,000 lb (71,000 kg) for both units. There will be three double-flow low-pressure

turbines, the same as those for the base case, 27 ft long by 15 ft high (8.2 by 4.6 m), at 1800 RPM and generating 171 MWe. The weight estimate for these three turbines is 1.11 x 10^6 lb (503,000 kg). The total weight of all potassium turbines for this case is 1.27 million lb (576,000 kg). The total cost of all turbines with generators is estimated to be $27.4 million. This lower cost was a consequence of the lower volume flows which permitted smaller machinery.

Case 11. For Case 11 the turbine inlet temperature is 1500 F (1089 K), which is intermediate between the temperatures of Case 10 and the rest of the cases.

The turbines for this case are similar to the designs in Table 10 of Reference 1 (p. 25), with the high-pressure turbine scaled to 3600 RPM (1/2 size) and the low-pressure turbine scaled to 1800 RPM (2/3 size). Using the flow rate of 8.9 x 10^6 lb/hr (67,000 kg/sec) from the performance calculation, it was determined that six double-flow high-pressure turbines and six double-flow low-pressure turbines are required. The six high-pressure turbines are 1/2 scale of Figure 3 of Reference 1 (p. 23). Each turbine is 16 ft long and 9 ft high (4.9 and 2.7 m). The total weight of all six is 471,000 lb (213,000 kg). The total cost (including two generators) is $15.9 million. The six low-pressure turbines are 2/3 scale of Figure 4 of Reference 1 (p. 27). Each turbine is 27 ft long and 15 ft high (8.2 and 4.6 m). Total weight of all six is 2 x 10^6 lb (910,000 kg).

The cost of the six low-pressure turbines (with two generators) is $36.3 million. The high-pressure turbines generate 145 MW total, and the low-pressure turbines generate a total of 217 MW.

The total cost of turbines and generators for this case is estimated to be $52.2 million.

Case 17. Case 17 is the base case with cesium as a working fluid. Using the results of the cycle performance calculations, preliminary turbine flow path designs were made for the cesium turbines. It was determined that 3 stage turbines with 80 in. (2.0 m) diameter disks running at 1200 RPM were a feasible design. The stresses were calculated and materials were selected. The weights were calculated by scaling Figure 13 of Reference 1 (p. 52) by 2/3 and deleting one stage. The costs were estimated using the methods described for potassium turbines.

There are five double-flow cesium turbines each 25.5 ft long and 15 ft high (7.8 and 4.6 m), running at 1200 RPM and generating a total of 381 MWe. The estimated total turbine weight is 1.68 million lb (760,000 kg). The total cost of turbines and generators is estimated to be $33.4 million.

Case 18. Case 18 is another cesium case, with the same turbine inlet temperature as Case 17, but with a high cesium con-

densing temperature. The turbines are similar to those of Case 17, but only two stages are required. Preliminary flow path designs were made for 80-in. (2.0 m) disk diameter and 1200 RPM. Stresses were calculated, materials were selected, and weights and costs were estimated as described above.

This system has three double-flow cesium turbines each 24 ft long and 15 ft high (7.3 and 4.6 m), generating a total of 300 MW at 1200 RPM. The estimated total turbine weight is 858,000 lb (390,000 kg). The estimated total cost for the turbines plus a generator is $20.5 million.

Pumps and Dump Tanks

In Reference 1, costs were presented for potassium components including boilers, turbines, condensers, pumps, and dump tanks. For the alkali metal pumps and dump tanks, the number of these components has been scaled by the liquid volume flow rates. The results are summarized below for the cases for which turbine costs were estimated. Shown in successive lines are the fluid, the mass flow rate, the liquid density, the volume flow rate, the number of 4500 gal/min (0.284 m^3/sec) pumps required and the estimated costs. The last two lines indicate the number of dump tanks, the same size as those of Reference 1, and the estimated costs.

It is possible that larger and fewer pumps and dump tanks may be less expensive. These options could be considered in further studies.

Liquid Metal Pump and Dump Tank Costs

Case	1-9,12-16	10	11	17	18
Fluid	K	K	K	Cs	Cs
Flow Rate, millions lb/hr	9.25	5.16	8.9	35.0	37.0
Density, lb/ft^3	44.	44.	44.	97.	95.
Flow Rate, gal/min	26210	14620	25220	44985	48560
No. Pumps	6	3	6	10	11
Total Cost, $ millions	1.782	0.891	1.782	2.97	3.267
No. Dump Tanks	6	3	6	10	11
Total Cost, $ millions	7.128	3.564	7.128	11.88	13.07

RESULTS

Results for the study of liquid metal topping cycles are tabulated in Table 2.7-1, which includes the major cycle input parameters.

Capital cost distributions are given in Table 2.7-2.

Summaries giving major cycle characteristics for the two base cases (Case 1 and Case 17) are given in Table 2.7-3 and 2.7-4.

Auxiliary losses and power outputs are shown in Table 2.7-5.

Table 2.7-1

PARAMETRIC VARIATIONS FOR TASK I STUDY LIQUID METAL TOPPING CYCLE

Parameters	Case 1*	2	3	4	5	6	7	8	9	10	11	12	13	16	17	18
Power Output (MWe)	1088	1081	1070	2433	2548	2594	1508	1613	1452	908	1077	1086	1071	1072	1129	1142
Furnace, Coal, and Conversion Process	AFB Ill. #6	AFB Mont	AFB N.D.	PF Ill. #6 LBtu	PF Mont LBtu	PF N.D. LBtu	PF Ill. #6 HBtu	AFB Ill. #6	(PFB)R Ill. #6	PF Ill. #6 HBtu	AFB Ill. #6 →					→
Fluidized Bed Temperature (°F)	1550 →		→	--	--	--	--	1650	1650	--	1550 →					→
Topping Cycle																
Fluid	K →													→	Cs →	→
Turbine inlet temperature (°F)	1400 →								→	1700	1500	1400 →				→
Regenerative feed heaters	0 →										→	1	0 →			→
Condensing temperature (°F)	1100 →													→	1000	1100
Bottoming Cycle																
Turbine inlet temperature (°F)	1000 →											→	950 →		→	1000
Maximum pressure (psig)	3500 →															→
Reheat temperature (°F)	1000 →											→	950 →		→	1000
Condensing pressure (in. Hg abs)	1.5 →												→	1.0	1.5 →	→
Maximum feedwater temperature (°F)	510 →															→
Heat rejection	WCT →												→	DCT	WCT →	→
Pressurized Furnace																
Percent excess air	--	--	--	15 →		→	10	--	20	10	--	--	--	--	--	--
Pressure ratio	--	--	--	10 →		→	8	--	10	8	--	--	--	--	--	--
Turbine inlet temperature (°F)	--	--	--	1800 →		→	1750	--	1800	1750	--	--	--	--	--	--
Regenerator efficiency	--	--	--	Steam for gasifier →		→	0.85	--	0.85 →	→	--	--	--	--	--	--
Actual Powerplant Output (MWe)	1088	1081	1070	2433	2548	2594	1508	1613	1452	908	1077	1086	1071	1072	1129	1142
Thermodynamic Efficiency (percent)	51.4	51.4	51.4	51.4	51.4	51.4	51.4	50.9	51.4	55.5	52.4	51.5	50.1	50.0	52.2	51.3
Powerplant Efficiency (percent)	38.9	36.9	33.9	35.1	35.5	35.2	40.6	38.5	39.6	43.3	40.0	38.6	37.7	37.1	41.5	40.8
Overall Energy Efficiency (percent)	38.9	36.9	33.9	35.1	35.5	35.2	20.5	38.5	39.6	21.8	40.0	38.6	37.7	37.1	41.5	40.8
Coal Consumption (lb/kWh)	0.81	1.03	1.46	0.90	1.08	1.41	1.55	0.82	0.80	1.45	0.79	0.82	0.84	0.85	0.76	0.78
Plant Capital Cost ($ million)	1280	1282	1292	2063	2155	2296	1021	1901	1332	831	1669	1269	1287	1391	1216	1219
Plant Capital Cost ($/kWe)	1176	1185	1207	848	845	885	677	1178	917	914	1548	1168	1202	1298	1076	1067
Cost of Electricity, Capacity Factor = 0.65																
Capital (mills/kWh)	37.[illegible]	37.5	38.2	26.8	26.7	28.0	21.4	37.3	29.0	28.9	48.9	36.9	38.0	41.0	34.0	33.7
Fuel (mills/kWh)	7.5	7.9	8.6	8.3	8.2	8.2	21.9	7.5	7.3	20.5	7.3	7.5	7.7	7.8	7.0	7.1
Maintenance and operating (mills/kWh)	3.7	4.0	4.1	3.4	3.4	3.6	2.1	3.2	3.3	3.1	5.3	3.9	4.1	4.1	3.6	3.6
Total (mills/kWh)	48.3	49.3	50.8	38.5	38.3	39.8	45.3	48.0	39.6	52.5	61.5	48.4	49.8	53.0	44.6	44.4
Sensitivity																
Capacity factor = 0.50 (total mills/kWh)	60.6	61.8	63.5	47.6	47.4	49.3	52.4	60.2	49.3	62.1	77.7	60.6	62.4	66.5	55.9	55.6
Capacity factor = 0.80 (total mills/kWh)	40.7	41.6	42.9	32.8	32.7	33.9	40.9	40.4	33.6	46.5	51.3	40.7	41.9	44.5	37.6	37.4
Capital Δ = 20 percent (Δ mills/kWh)	7.4	7.5	7.6	5.4	5.3	5.6	4.3	7.5	5.8	5.8	9.8	7.4	7.6	8.2	6.8	6.7
Fuel Δ = 20 percent (Δ mills/kWh)	1.5	1.6	1.7	1.7	1.6	1.6	4.4	1.5	1.5	4.1	1.5	1.5	1.5	1.6	1.4	1.4
Estimated Time for Construction (years)	6	6	6	6	6	6	6	6	6	6	6	6	6	6	6	6
Estimated Date of 1st Commercial Service (year)	1992	1992	1992	1992	1992	1992	1992	1992	1992	1998	1994	1992	1992	1992	1995	1995

*Base case.

AFB = Atmospheric fluidized bed
DCT = Dry cooling tower
HBtu = High Btu
Ill. = Illinois
LBtu = Low Btu
Mont = Montana
N.D. = North Dakota
PF = Pressurized furnace
$(PFB)_R$ = Pressurized fluidized bed (recuperative)
WCT = Wet cooling tower

Table 2.7-2 (Page 1 of 2)

CAPITAL COST DISTRIBUTIONS FOR LIQUID METAL TOPPING CYCLE

	CASE NO.	1	2	3	4	5	6	7	8	9	10
MAJOR COMPONENTS											
PRIME CYCLE											
LIQUID METAL TURB-GEN	MM$	43.3	43.3	43.3	43.3	43.3	43.3	43.3	65.0	43.3	27.4
LIQUID METAL PUMP	MM$	1.8	1.8	1.8	1.8	1.8	1.8	1.8	2.7	1.8	0.9
LIQUID METAL DUMP TANK	MM$	7.1	7.1	7.1	7.1	7.1	7.1	7.1	10.7	7.1	3.6
BOTTOMING CYCLE											
CONDENSER-BOILER	MM$	2.6	3.0	2.6	2.8	2.8	2.8	2.4	3.9	3.2	1.4
STEAM TURB-GEN	MM$	30.2	30.2	30.2	30.2	30.2	30.2	30.2	39.5	30.2	19.2
PRIMARY HEAT INPUT AND FUEL SYSTEM											
FURNACE MODULES	MM$	212.9	214.1	221.1	76.3	76.0	79.7	96.9	320.7	209.4	48.8
HIGH TEMP AIR PREHEATER	MM$	25.1	22.3	20.6	0.	0.	0.	0.	33.1	0.	0.
LOW TEMP AIR PREHEATER	MM$	3.9	1.7	2.3	0.	0.	0.	0.	5.5	0.	0.
PRESSURIZING GAS TURBINE (COMP-GEN-HEAT EXCH)	MM$	0.	0.	0.	123.4	129.6	134.1	37.6	0.	54.1	21.3
GASIFIER (INCLUDING BOOST STEAM TURB-COMP	MM$	0.	0.	0.	403.2	437.8	488.1	0.	0.	0.	0.
SUB-TOTAL OF MAJOR COMPONENTS	MM$	326.9	323.5	329.0	688.1	728.6	787.2	219.4	481.1	349.1	122.6
BALANCE OF PLANT											
COOLING TOWER	MM$	6.2	6.2	6.2	6.2	6.2	6.2	6.2	9.3	6.2	6.2
ALL OTHER	MM$	247.4	250.4	250.4	270.1	274.4	283.8	229.9	371.2	251.0	229.9
SITE LABOR	MM$	70.5	71.8	71.8	85.0	87.0	91.0	63.8	105.7	71.3	63.8
SUB-TOTAL OF BALANCE OF PLANT	MM$	324.1	328.4	328.4	361.2	367.6	381.0	299.9	486.2	328.5	299.9
CONTINGENCY	MM$	130.2	130.4	131.5	209.9	219.2	233.6	103.9	193.4	135.5	84.5
ESCALATION COSTS	MM$	223.4	223.7	225.6	360.1	376.2	400.9	178.2	331.9	232.5	145.0
INTEREST DURING CONSTRUCTION	MM$	275.2	275.6	277.9	443.6	463.4	493.8	219.5	408.9	286.4	178.6
TOTAL CAPITAL COST	MM$	1279.6	1281.6	1292.4	2062.9	2155.0	2296.4	1020.8	1901.5	1332.0	830.5
MAJOR COMPONENTS COST	$/KWE	300.5	299.4	307.3	282.9	286.0	303.4	145.5	298.3	240.4	135.0
BALANCE OF PLANT	$/KWE	297.9	303.9	306.8	148.5	144.3	146.9	198.9	301.4	226.3	330.3
CONTINGENCY	$/KWE	119.7	120.7	122.8	86.3	86.0	90.1	68.9	119.9	93.3	93.0
ESCALATION COSTS	$/KWE	205.3	207.0	210.7	148.0	147.6	154.5	118.2	205.8	160.1	159.6
INTEREST DURING CONSTRUCTION	$/KWE	252.9	256.0	259.6	182.4	181.9	190.4	145.6	253.5	197.3	196.7
TOTAL CAPITAL COST	$/KWE	1176.4	1186.0	1207.3	848.1	845.8	885.3	677.1	1178.9	917.4	914.6

Table 2.7-2 (Page 2 of 2)

CAPITAL COST DISTRIBUTIONS FOR LIQUID METAL TOPPING CYCLE

	CASE NO.	11	12	13	16	17	18
MAJOR COMPONENTS							
PRIME CYCLE							
LIQUID METAL TURB-GEN	MM$	52.2	43.3	43.3	43.3	33.4	20.5
LIQUID METAL PUMP	MM$	1.8	1.9	1.8	1.8	3.0	3.3
LIQUID METAL DUMP TANK	MM$	7.1	7.5	7.3	7.4	11.9	13.1
BOTTOMING CYCLE							
CONDENSER-BOILER	MM$	2.4	2.6	2.3	2.7	3.4	2.6
STEAM TURB-GEN	MM$	24.7	30.2	29.7	30.2	24.2	30.3
PRIMARY HEAT INPUT AND FUEL SYSTEM							
FURNACE MODULES	MM$	409.8	205.2	216.9	220.2	171.9	178.6
HIGH TEMP AIR PREHEATER	MM$	22.5	27.4	25.5	26.0	24.2	25.1
LOW TEMP AIR PREHEATER	MM$	3.7	3.2	3.9	4.0	3.7	3.9
PRESSURIZING GAS TURBINE (COMP-GEN-HEAT EXCH)	MM$	0.	0.	0.	0.	0.	0.
GASIFIER (INCLUDING BOOST STEAM TURB-COMP	MM$	0.	0.	0.	0.	0.	0.
SUB-TOTAL OF MAJOR COMPONENTS	MM$	524.3	321.2	330.7	335.6	275.6	277.3
BALANCE OF PLANT							
COOLING TOWER	MM$	6.2	6.2	6.2	24.7	6.2	6.2
ALL OTHER	MM$	247.4	247.4	247.4	263.2	262.4	262.4
SITE LABOR	MM$	70.5	70.5	70.5	84.4	74.2	74.2
SUB-TOTAL OF BALANCE OF PLANT	MM$	324.1	324.1	324.1	372.2	342.7	342.7
CONTINGENCY	MM$	169.7	129.1	131.0	141.6	123.7	124.0
ESCALATION COSTS	MM$	[illegible]	221.4	224.7	242.9	212.2	212.8
INTEREST DURING CONSTRUCTION	MM$	358.6	272.8	276.8	299.2	261.4	262.1
TOTAL CAPITAL COST	MM$	1667.7	1268.5	1287.3	1391.5	1215.6	1218.9
MAJOR COMPONENTS COST	$/KWE	486.7	295.8	308.8	313.1	244.2	242.9
BALANCE OF PLANT	$/KWE	300.8	298.4	302.6	347.3	303.6	300.2
CONTINGENCY	$/KWE	157.5	118.8	122.3	132.1	109.6	108.6
ESCALATION COSTS	$/KWE	270.2	203.9	209.8	226.6	188.0	186.3
INTEREST DURING CONSTRUCTION	$/KWE	332.9	251.2	258.5	279.2	231.6	229.5
TOTAL CAPITAL COST	$/KWE	1548.1	1168.1	1202.1	1298.3	1076.9	1067.5

Table 2.7-3

SUMMARY SHEET
LIQUID METAL TOPPING CYCLE BASE CASE NO. 1

CYCLE PARAMETER

Net Power Output (MWe)	1088
Furnace and Coal Type	Atmospheric fluidized bed Illinois No. 6
Prime Cycle	
Fluidized bed temperature (°F)	1550
Fluid	Potassium
Turbine inlet temperature (°F)	1400
Condensing temperature (°F)	1100
Bottoming Cycle	
Turbine inlet temperature (°F)	1000
Maximum pressure (psig)	3500
Reheat temperature (°F)	1000
Condensing pressure (in. Hg abs.)	1.5
Heat Rejection	Wet cooling tower

PERFORMANCE AND COST

Thermodynamic efficiency (percent)	51.4
Powerplant efficiency (percent)	38.9
Overall energy efficiency (percent)	38.9
Plant capital cost ($ x 10^6)	1071
Plant capital cost ($/kWe)	984
Cost of electricity (mills/kWh)	41.0

NATURAL RESOURCES

Coal (lb/kWh)	0.813
Water (gal/kWh)	
Total	0.79
Cooling	0.79
Processing	0
Makeup	0
NO_x suppression	0
Stack gas cleanup	0
Land (acres/100 MWe)	4.6

ENVIRONMENTAL INTRUSION

	Lb/10^6-Btu Input	Lb/kWh Output
SO_2	1.26	0.0096
NO_x	0.319	0.0024
HC	0	0
CO	0.249	0.0019
Particulates	0.1	7.5×10^{-4}

	Btu/kWh
Heat to water	3497
Heat, total rejected	5359

	Lb/kWh	Lb/Day
Wastes		
Furnace solids	0.143	3.74×10^6
Fine dust from cyclones	0.109	2.84×10^6
Fly ash	0.015	0.39×10^6

MAJOR COMPONENT CHARACTERISTICS

Major Component	Unit or Module: Size (ft) (W x L (or D) x H)	Weight (lb) (x 10^6)	Cost ($ x 10^6)	Units Required	Total Cost ($ x 10^6)	$/kW Output
Prime Cycle						
Metal vapor turbine-generator	15 x 120 x 15	1.31	21.65	2	43.3	39.8
Liquid metal dump tanks	12 x 12 x 50	0.3	1.19	6	7.1	6.5
Bottoming Cycle						
Condenser-boiler	15 x 20 x 130	0.20	0.478	5.50	2.4	2.2
Steam turbine-generator	30 x 198 x 25	6.3	30.25	1	30.25	27.8
Primary Heat Input System						
Furnace module	12.5 x 31 x 330	5.3	38.73	5.5	213.0	195.8
High-temperature air preheater	35 x 46 x 4	0.11	0.304	82.5	25.1	23.1

Table 2.7-4

SUMMARY SHEET
LIQUID METAL TOPPING CYCLE BASE CASE NO. 17

CYCLE PARAMETER

Net Power Output (MWe)	1129
Furnace and Coal Type	Atmospheric fluidized bed Illinois No. 6
Prime cycle	
Fluidized bed temperature ($^{\circ}F$)	1550
Fluid	Cesium
Turbine inlet temperature ($^{\circ}F$)	1400
Condensing temperature ($^{\circ}F$)	1000
Bottoming Cycle	
Turbine inlet temperature ($^{\circ}F$)	950
Maximum pressure (psig)	3500
Reheat temperature ($^{\circ}F$)	950
Condensing pressure (in. Hg abs.)	1.5
Heat Rejection	Wet cooling tower

PERFORMANCE AND COST

Thermodynamic efficiency (percent)	52.2
Powerplant efficiency (percent)	41.5
Overall energy efficiency (percent)	41.5
Plant capital cost ($ x 10^6)	1216
Plant capital cost ($/kWe)	1076
Cost of electricity (mills/kWh)	44.6

NATURAL RESOURCES

Coal (lb/kWh)	0.859
Water (gal/kWh)	
Total	0.76
Cooling	0.76
Processing	0
Makeup	0
NO_x suppression	0
Stack gas cleanup	0
Land (acres/100 MWe)	4.4

ENVIRONMENTAL INTRUSION

	Lb/10^6-Btu Input	Lb/kWh Output
SO_2	1.26	0.0089
NO_x	0.319	0.0022
HC	0.0	0.0
CO	0.249	0.0018
Particulates	0.1	7.0×10^{-4}

	Btu/kWh
Heat to water	3463
Heat, total rejected	4810

	Lb/kWh	Lb/Day
Wastes		
Furnace solids	0.134	3.62×10^6
Fine dust from cyclones	0.102	2.76×10^6
Fly ash	0.014	0.38×10^6

MAJOR COMPONENT CHARACTERISTICS

Major Component	Unit or Module: Size (ft) (W x L (or D) x H)	Weight (lb) (x 10^6)	Cost ($ x 10^6)	Units Required	Total Cost ($ x 10^6)	$/kW Output
Prime Cycle						
Metal vapor turbine-generator	15 x 105 x 15	1.09	16.7	2	33.4	29.6
Liquid metal dump tanks	12 x 50 x 12	0.3	1.19	10	11.9	10.5
Bottoming Cycle						
Condenser-boiler	[illegible] x 20 x 173	0.26	3.4	1	3.4	3.0
Steam turbine-generator	30 x 174 x 25	5	24.2	1	24.2	21.4
Primary Heat Input System						
Furnace module	12.5 x 31 x 330	7.0	32.40	5.3	17.19	152.3
High-temperature air preheater	35 x [illegible] x 4	0.11	0.304	80.1	24.2	21.4

Table 2.7-5

POWER OUTPUT AND AUXILIARY POWER DEMAND FOR BASE CASE AND PARAMETRIC VARIATIONS: LIQUID METAL TOPPING CYCLE

CASE NO.		1	2	3	4	5	6	7	8	9	10
PRIME CYCLE POWER OUTPUT	MW	298.4	298.4	298.4	298.4	298.4	298.4	298.4	447.7	298.4	282.7
BOTTOMING CYCLE POWER OUTPUT	MW	941.9	941.9	941.9	941.9	941.9	941.9	941.9	1397.0	941.9	469.3
FURNACE POWER OUTPUT	MW	0.	0.	0.	1293.0	1408.0	1460.0	351.0	0.	324.5	197.2
BALANCE OF PLANT AUX. POWER REQ'D.	MW	29.0	29.2	29.4	29.0	29.2	29.4	28.4	42.6	29.0	28.4
FURNACE AUX. POWER REQ'D.	MW	117.3	124.3	134.2	59.1	57.8	63.3	47.3	[illegible]	76.1	8.0
TRANSFORMER LOSSES	MW	6.2	6.2	6.2	12.7	13.2	13.5	8.0	9.2	7.8	4.7
NET STATION OUTPUT	MW	1087.8	1080.6	1070.5	2432.5	2548.1	2594.1	1507.6	1612.9	1451.9	908.1

CASE NO.		11	12	13	16	17	18
PRIME CYCLE POWER OUTPUT	MW	361.7	301.8	302.8	307.9	381.1	296.3
BOTTOMING CYCLE POWER OUTPUT	MW	861.5	940.0	926.2	938.9	844.5	944.1
FURNACE POWER OUTPUT	MW	0.	0.	0.	0.	0.	0.
BALANCE OF PLANT AUX. POWER REQ'D.	MW	29.0	29.0	29.0	44.6	29.0	29.0
FURNACE AUX. POWER REQ'D.	MW	110.8	120.6	123.0	124.2	61.7	63.4
TRANSFORMER LOSSES	MW	6.1	6.2	6.1	6.2	6.1	6.2
NET STATION OUTPUT	MW	1077.3	1086.0	1070.9	1071.8	1128.8	1141.8

DISCUSSION OF RESULTS

Major Characteristics

The results shown in Table 2.7-1 show relatively good efficiencies; the overall energy efficiency was as high as 40.0 percent for potassium and 41.5 percent for cesium. As a consequence of the high efficiency, coal consumption was relatively low-as low as 0.80 pound of coal per kWh (100 kg/GJ).

Offsetting these advantages is a relatively high cost of electricity (48.3 and 44.6 mills/kWh for the potassium and cesium base cases, respectively). This high cost of electricity is largely a consequence of the high capital costs ($1176/kW for the potassium base case). A major contribution to capital costs was made by the liquid metal boilers and auxiliaries (a total of $196/kW for the potassium base case). The construction time, estimated to be six years, also contributed heavily to capital cost; interest and escalation during construction totaled $458/kW for the same case. Balance-of-plant costs were $298/kW for that case. Balance-of-plant costs were high because of the need for safety provisions and a large quantity of high-temperature piping.

Discussion

A number of observations can be made from the results shown in Table 2.7-1.

Case 12 shows that a regenerative feed heater in the potassium circuit offered no advantage over the potassium base case, which had no feed heating. There also was no advantage to lowering the heat input temperatures to the bottoming cycle (Case 13).

The higher temperature pressurized furnace case (Case 11) offered no advantage in cost of electricity over the lower temperature pressurized furnace case (Case 7). Likewise, the higher temperature atmospheric fluidized bed case (Case 11) produced more costly power than the lower temperature atmospheric fluidized bed cases. The lower temperature difference between the bed and the potassium in Case 11 was largely responsible for the increased cost.

It can be seen in Table 2.7-1 that a low cost of electricity was produced by those cases with pressurized furnaces using low-Btu gas (Cases 4, 5, and 6). In these cases, however, about one-half the total plant power was produced by the expansion turbine downstream of the pressurized furnace. Thus, this system is thermodynamically equivalent to a gas turbine in parallel with a liquid metal topping system. The decrease in cost of electricity is primarily due to the lower capital cost associated with gas turbine systems, and does not reflect the merits of the liquid metal topping system. This system does not appear to warrant further study, as the benefit of the cycle appears to result

C2

primarily from the gas turbine. Moreover, the overall energy efficiency was not favorable.

The pressurized fluidized bed case (Case 9) showed the lowest cost of electricity, and in that case the power produced by the expansion turbine was 21 percent of the total plant power. However, the potential for pressurized fluidized bed power generation in power generation cycles is still unproven because of the possibility of hot corrosion in the gas turbine that must operate on the products of coal combustion. The use of the turbine exhaust for feedwater heating in the bottoming system was not studied. Instead, it was decided for analytical convenience to use gas turbine regeneration so that the furnace system would be decoupled from the topping and bottoming cycles. The pressurized fluidized bed with gas turbine regeneration can be compared with the same type furnace with gas turbine exhaust heating feedwater, by referring to Section 2.6, "Advanced Steam Cycle." This comparison shows that heating feedwater produces an increase in efficiency at the penalty of higher cost of electricity.

The use of high-Btu gas in a pressurized furnace (Cases 7 and 10) resulted in a very low overall energy efficiency as a consequence of the low coal conversion efficiency.

The cesium cases (Cases 17 and 18) showed improvements in efficiency and cost of electricity compared with the equivalent potassium cases. The primary reason is the higher density of cesium, resulting in smaller components and more efficient heat transfer. In addition, the boiler recirculation power requirement is lower than for potassium. While cesium does have these advantages, there are some questions regarding the corrosiveness of cesium. Furthermore, most of the development of liquid metal power systems has been done so far with potassium. The use of cesium in this application is considered speculative at this time.

One of the large capital cost and power consuming components of the furnace system is the liquid metal recirculating pump. When the Foster Wheeler Energy Corporation designed the boiler, the assumption was made that a high mass flux had to be maintained within the boiler tubes to prevent hot spots or burnout. In addition, a large entrance flow restriction was used to assure uniform flow distribution. For the potassium base case, the recirculation pumps added a capital cost of $36.4 million, and consumed 57 MW of power.

It is probable that further boiler design studies could reduce, or even eliminate, the need for recirculation. It is estimated that if the power and capital cost of recirculation could be reduced to one-half the present levels, the potassium base case (Case 1) cost of electricity could be reduced by 1.8 mills/kWh, and the overall efficiency could be increased by 1.0 percent. The effect of other degrees of recirculation may be calculated from these figures by proportion. For example, if recirculation were eliminated entirely, the above figures would approximately double.

RECOMMENDED CASE

The disadvantages or uncertainties of cesium, pressurized fluidized beds, temperatures higher than 1400 F (1033 K), and pressurized furnaces were described above. The low overall energy efficiency of high-Btu gas discourages further consideration. The cases that overcome the above objections are the atmospheric fluidized bed cases. Of these cases, the potassium base case (Case 1) is the one that is considered best for further study. The major parameters for that case are shown in Table 2.7-1.

REFERENCES

1. Rossbach, R.J., Final Report of Joint NASA/OCR Study of Potassium Topping Cycles for Stationary Power, NASA Contract NAS3-17354, Space Division, General Electric Company, Cincinnati, Ohio 45215, November 13, 1973.

2. Ewing, C.T., et al., High Temperature Properties of Potassium, Report NRL-6233, Naval Research Laboratory, Washington, D.C., September 24, 1965.

3. Ewing, C.T., et al., High Temperature Properties of Cesium, Report NRL-6246, Naval Research Laboratory, Washington, D.C., September 24, 1965.

4. Ainley, D.G., and Mathieson, G.C.R., A Method of Performance Estimation for Axial Flow Turbines, R&M 2974, Aeronautical Research Council (Great Britain), Her Majesty's Stationery Office, London, England, 1951.

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